DOE/NASA/0374 - 1
NASA CR - 187158

Development of Advanced High Temperature In-Cylinder Components and Tribological Systems For Low Heat Rejection Diesel Engines.

Phase 1 - Final Report

C.A. Kroeger
H.J. Larson

March 1992

Prepared For
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract DEN3-374

for
U.S. DEPARTMENT OF ENERGY
Conservation and Renewable Energy
Office of Propulsion Systems

TABLE OF CONTENTS

GLOSSARY OF ACRONYMS

BMEP - brake mean effective pressure

BSFC - brake specific fuel consumption

CR - compression ration

FE - finite element

FMEP - friction mean effective pressure

MPG - miles per gallon

PCP - peak cylinder pressure

POF - probability of failure

RBC - Rankine bottoming cycle

ROI - return on investment

SFC - specific fuel consumption

TBC - thermal barrier coating

TRG - top ring groove

TRR - top ring reversal

WRCAS - whisker reinforced calcium-alumino-silicate

3DFEA - 3 dimensional finite element analysis

LIST OF FIGURES

LIST OF FIGURES (Cont'd)

LIST OF TABLES

LIST OF TABLES (Cont'd)

SUMMARY

This report covers the results of Phase I, of a three phase contract, to develop in-cylinder components and tribological systems for low heat rejection diesel engines with a specific fuel consumption goal of 152 g/kW-hr (0.25 lbs./BHP-hr). The objective of Phase I was to select in-cylinder components and tribological systems for design and bench test evaluation in Phase II of the contract. Phase III provides for in-cylinder component evaluation in a single cylinder engine.

Three concept engine configurations were evaluated in Phase I. The concept engines were based on in-line, six cylinder, four stroke cycle, low heat rejection components. Water cooling in the engines was eliminated by using selective oil cooling of the in-cylinder components. A conventional lubrication and bearing system was retained, but a high temperature lubricant will be needed to meet the higher in-cylinder operating temperatures. The concept engines incorporated exhaust energy recovery systems.

The Phase I study concluded that the specific fuel consumption goal was feasible, if the engine and in-cylinder components were designed for cylinder pressures in the 22 to 24 MPa (3200 psi to 3500 psi) range. Preliminary design and analysis of in-cylinder components, at the proposed operating conditions, were completed to evaluate the in-cylinder component life potential. A turbo-compound system for exhaust energy recovery was incorporated to meet the specific fuel consumption goal.

The proposed low heat rejection, in-cylinder component concepts and high efficiency turbomachinery are expected to increase the concept engine cost by 43%. Higher initial engine cost will be offset by reduced operating costs due to lower fuel consumption. An economic analysis, based on customer owning and operating costs, indicated a return on investment (ROI) ranging from 32 to 36% at a diesel fuel price of $1.00 per gallon.

I INTRODUCTION

The objective of the DOE/NASA Heavy Duty Transport Technology program is to develop a technology base that can be applied to the design of fuel efficient, low heat rejection heavy duty diesel engines with a brake specific fuel consumption of 0.25 #/BHP-hr (152 g/kW-hr). These engines must be cost effective, meet future emission and environmental goals, and have the potential for rapid technology transfer to the design of heavy duty truck engines for the early 2000's.

The objective of this contract (DEN3-374) is to evaluate advanced in-cylinder components and tribological systems for high temperature, low heat rejection heavy duty truck diesel engines. The contract is divided into three phases. Phase 1 provides for the concept selection of in-cylinder components and tribological systems that will be needed to meet the fuel consumption, durability, cost and reliability goals. Phase II covers the in-cylinder components concept design and bench test evaluation. Phase III is the concept verification phase to demonstrate in-cylinder component feasibility in a single cylinder engine.

The Phase I work plan included three tasks;
- Task 1 - Analysis of proposed engine concepts,
- Task 2 - Component concept selection,
- Task 3 - Phase I briefing.

Task 1 was broken down into the following elements;
1) Engine system concepts,
2) In-cylinder component concepts,
3) Tribology concepts,
4) Preliminary design and review.

II - ENGINE SYSTEM CONCEPTS

Engine Functional Specifications

The first step in the Task 1 concept analysis was to formulate an engine functional specification that would serve as a guide for evaluating proposed engine and in-cylinder component concepts. The engine functional specification shown in Table 1 reflects Caterpillar's view of the anticipated engine requirements for heavy duty truck engines of the 2000's.

TABLE 1 - HEAVY DUTY TRUCK ENGINE FUNCTIONAL SPECIFICATION

RATED POWER	186 - 298 kW (250 - 400 HP)
SPECIFIC FUEL CONSUMPTION (RATED POWER)	152 g/kW-hr (0.25 lb/HP-hr)
TORQUE RISE	> 35% at 66% speed
WEIGHT	< 908 kg (2000 lb)
SIZE LxWxH	< 1400 x 840 x 1150 mm (55" x 33" x 45")
DURABILITY	> 10,000 hr (B50) to overhaul
LUBRICANT CHANGE INTERVAL	53,000 km (20,000 miles)
ALTITUDE CAPABILITY	2300 meters (7500 feet)
NOISE LEVEL	<94.5 dB_a at 1 meter (SAE 1074)
EMISSIONS (1994)	NO_x ≤ 5.0 g/HP-hr Particulates ≤ 0.1 g/HP-hr HC ≤ 1.3 g/HP-hr CO ≤ 15.5 g/HP-hr

The engine power ratings in the functional specification covers the range from 186 kW to 298 kW (250 to 400 horsepower). Caterpillar's view of opportunities in the Class 6 and 8 truck markets (19,501-80,000 lb. GVW) indicates that significant numbers of engines will continue to be needed in this power range.

The rated power specific fuel consumption goal of 152 g/kW-hr was used to evaluate and select the concept engine and in-cylinder component options. The part-load fuel consumption of the engine was also evaluated during the concept study. On-highway truck engines usually operate at less than rated power a significant portion of time. The truck fuel economy is determined by the part-load fuel consumption characteristic of the engine and the rated power performance. The concept engine should be optimized for a combination of part-load and rated load fuel consumption.

A desire to reduce the frequency of shifting gears on modern trucks leads to a driver preference for engines with a high torque rise and a wide speed range. The engine should have higher levels of torque at a lower percentage of rated engine speed. The engine functional specification includes a requirement for 35% torque rise at 66% of rated engine speed. Future engines must have these torque rise characteristics to remain competitive.

The concept engine weight and physical dimensions have been included as a goal in the functional specifications. These goals reflect current state-of-the-art diesel truck engines. Previous studies at Caterpillar have shown that a turbocompound engine can fit within this envelope.

The emissions limits shown in Table 1 are the 1994 Federal EPA standards for the transient test cycle. These emission limits are challenging for current diesel engines and will be even more challenging for insulated engines with higher cycle mean effective temperatures. The noise level goal is aggressive compared to today's engines but is low enough to allow vehicle manufacturers to meet anticipated pass-by noise standards.

Engine System Concept Analysis

The purpose of the engine system concept analysis was to evaluate the specific fuel consumption (SFC) sensitivity to a wide range of engine operating variables and component concept designs. The engine system analyses utilized a Caterpillar cycle simulation program to evaluate the options. The engine cycle simulation program was used to systematically evaluate a range of engine configurations, in-cylinder component options, and operating parameters.

The starting point for the concept engine analyses was the definition of an engine configuration that could be used as the reference for SFC sensitivity analyses. Previous experience with engines designed to achieve a SFC of 160 g/kW-hr indicated that peak cylinder pressures (PCP)above 19 MPa and brake mean effective pressures (BMEP) in the range of 2.5 - 3.5 MPa would be needed to meet the 152 g/kW-hr SFC goal. A 7 liter engine with a target power of 261 kW (350 hp) was selected for the initial concept studies.

The 7 liter engine configuration is described in Table 2. Piston and cylinder head concepts were assumed to have low heat rejection characteristics. The cylinder liner incorporated oil cooling near the top ring reversal region. The engine air system incorporated air-to-air charge cooling. A turbocompound stage was included after the turbochargers to recover energy from the exhaust system.

TABLE 2 - CONCEPT ENGINE CONFIGURATION

RATED POWER	261kW (350 HP)
SPEED	1600 RPM
DISPLACEMENT	7.0 LITER
BORE X STROKE	110 mm X 123 mm
CONFIGURATION	IN-LINE 6 CYLINDER
TYPE	FOUR STROKE CYCLE
FUEL SYSTEM	DIRECT INJECTION
AIR SYSTEM	TURBOCHARGED WITH AIR-TO-AIR CHARGE COOLING AND TURBOCOMPOUNDING
PISTON	INSULATED - HEAT REJECTION <1200 W
CYLINDER HEAD	FOUR VALVE UNCOOLED HEAD WITH CERAMIC BOTTOM DECK INSERT
CYLINDER LINER	CAST IRON WITH OIL COOLING NEAR TOP RING REVERSAL

A summary of the concept engine variables and levels investigated using the engine cycle simulation is shown in Table 3. The cycle simulation program is a numerical representation of the thermodynamic processes in the cylinder of a diesel engine. Accurate modeling of the in-cylinder conditions is supported by other engine sub-models such as component heat transfer, air flow through the ports and manifolds, and air system modeling using aerodynamic maps for turbomachinery. The simulation output provides overall engine performance, the state of the cylinder working fluid during the cycle, and the temperature and heat rejection information for the engine components.

TABLE 3 - CONCEPT ENGINE CYCLE VARIABLES

Variable	Levels Investigated
Peak Cylinder Pressure	19, 22, and 25 MPa
Brake Mean Effective Pressure	1.5 to 4.0 MPa (BMEP)
Compression Ratio	14:1 to 23:1
Stroke/Bore Ratio	1.0, 1.12 and 1.24
Component Insulation	Piston, head, liner, exhaust port and manifold
Valve Events	Overlap, maximum lift and timing
Turbocharger Efficiency	58 to 78%
Power Turbine Efficiency	80 to 90%
Exhaust/Intake Manifold Pressure Ratio	0.9 to 1.35
Compressor Pressure Ratio	2.5:1 (single) to 6.0:1 (series)
Intercooling - Aftercooling	$0^{o}C$ sub-ambient, $43^{o}C$ air-to-air, $140^{o}C$ (no aftercooling)
Friction/Parasitic Losses	0 to 120 kPa (FMEP)
Exhaust Energy Recovery	Turbocompounding Rankine Bottoming Cycle

Impact of Peak Cylinder Pressure on Engine SFC

The 7.0 liter engine, described in Table 2, was used to quantify the expected change in SFC as a function of peak cylinder pressure (PCP). The 7.0 liter engine was used throughout the concept engine performance analyses as the reference engine configuration.

The minimum SFC obtainable at a given cylinder pressure limit requires an optimum combination of air-fuel ratio, BMEP level, compression ratio (CR), and fuel injection timing. Experience with highly rated turbocompound engines has shown that best SFC can be obtained near an air-fuel ratio of 29:1 and a combustion duration of 40° (crankshaft angle degrees). In the results to be discussed, the air-fuel ratio was held constant at 29:1, with combustion duration (90% of the fuel burned) held at 40°.

The variation of SFC at a peak cylinder pressure of 19 MPa is shown as a function of compression ratio and BMEP in Figure 1. The air system assumed for the 19 MPa PCP engine was a single stage turbocharger with aftercooling and turbocompounding. The minimum SFC, at the 19 MPa cylinder pressure limit, is 161 g/kW-hr with a BMEP range of 2-2.25 MPa. The 19 MPa engine does not approach the 152 g/kW-hr goal, unless a Rankine bottoming cycle (RBC) is added to the engine configuration.

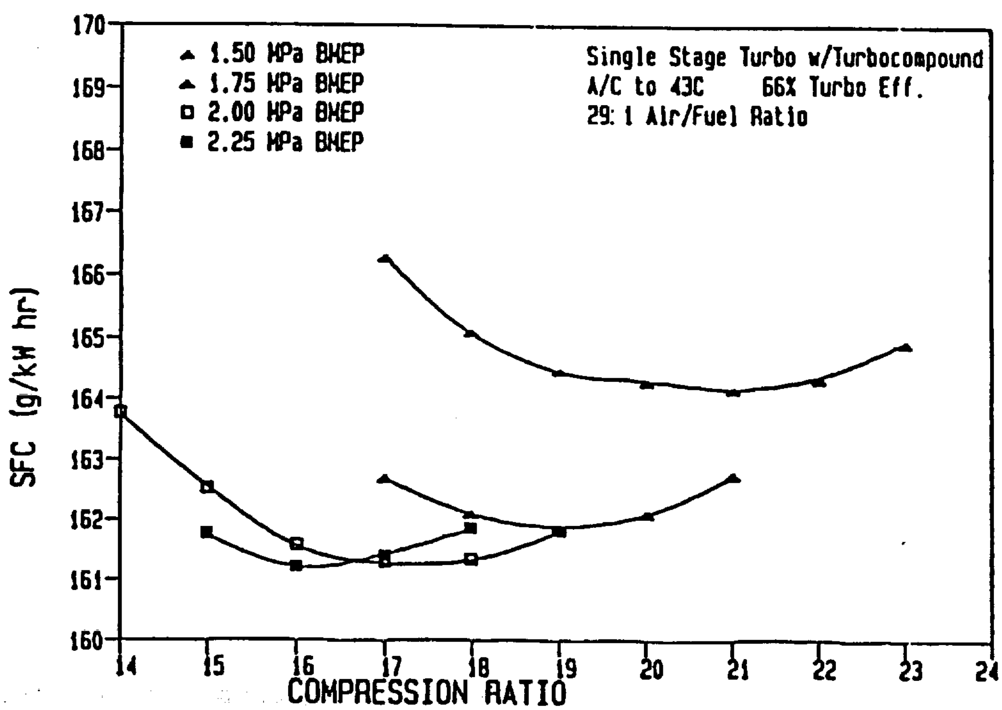


Figure 1 - SFC as a Function of BMEP and CR at 19 MPa PCP

The SFC sensitivity to BMEP and compression ratio for 22 and 25 MPa cylinder pressure limits is shown in Figure 2. Series turbochargers were used at 22 and 25 MPa to meet the higher boost pressures required to obtain a 29:1 air-fuel ratio at the higher BMEP's.

Intercooling between the low and high pressure turbocharger compressors was included in both the 22 and 25 MPa engine cycle simulations. The assumed turbocharger efficiency was increased from 70 % to 74% for the series turbocharged engines. Higher compressor stage efficiencies are achievable with the 2:1 compressor pressure ratio, series turbo stages, compared to the 3:1 compression ratio for the single stage turbocharger system.

At 22 MPa PCP, a minimum SFC of 155.0 g/kW-hr is predicted at 3.5 MPa BMEP. A minimum SFC of 152.6 g/kW-hr has been predicted for the 25 MPa PCP case. Both Figures 1 and 2 indicate that the variation in SFC with BMEP is small near the minimum SFC point.

The SFC sensitivity to cylinder pressure and the air system configuration is summarized in Figure 3. The 152 g/kW-hr SFC goal is predicted to be feasible at 24 to 25 MPa peak PCP.

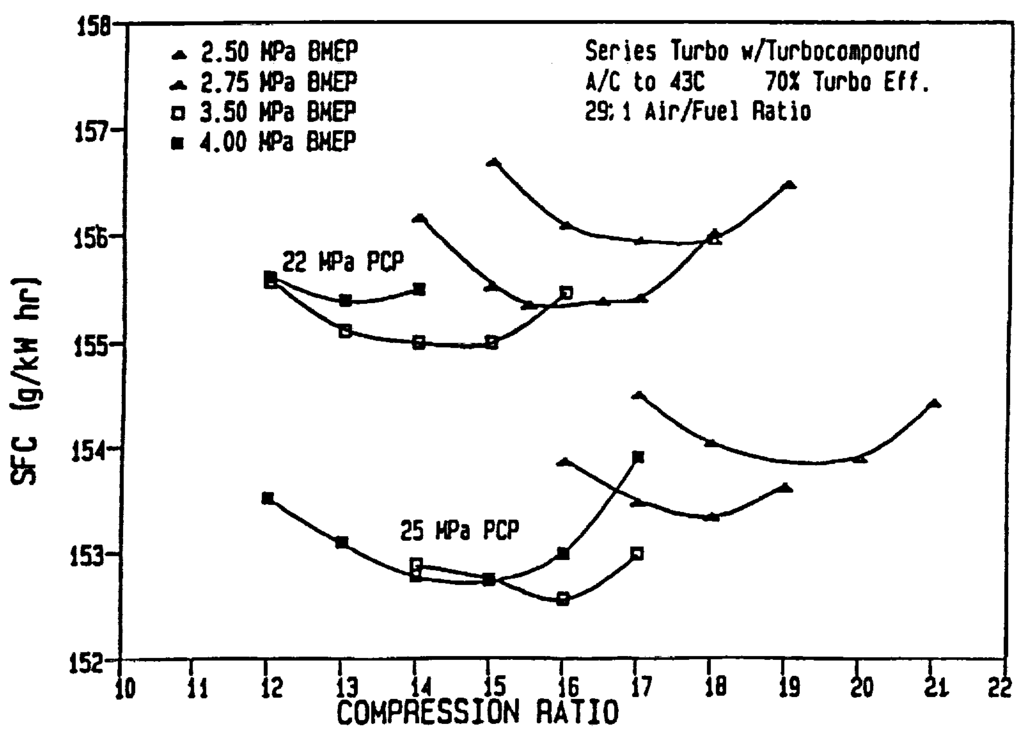


Figure 2 - SFC Sensitivity to BMEP, CR and Cylinder Pressure

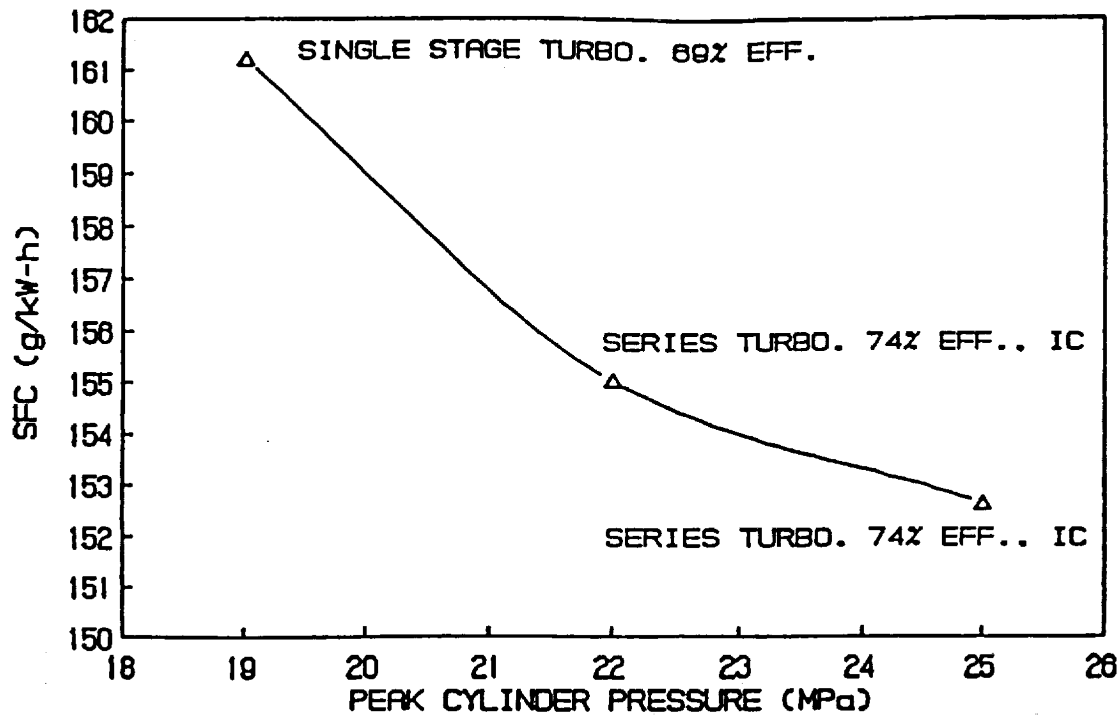


Figure 3 - SFC Sensitivity to Cylinder Pressure and Air System

SFC Sensitivity to Stroke/Bore Ratio

The engine operating conditions selected to evaluate stroke-bore ratio and component heat rejection with the baseline 7.0 liter engine geometry were: 261 kW rated power, 22 MPa peak cylinder pressure, 2.8 MPa BMEP, and 1600 rpm. This engine configuration served as the baseline engine for the sensitivity work in the concept engine analyses.

Variations in stroke-bore ratio can have an impact on engine fuel consumption due to tradeoffs in the design of in-cylinder components. The stroke-bore ratio changes, 1) the amount of heat transfer surface area, 2)cylinder head area available for valves, 3) the shape of the combustion chamber, and 4) the maximum piston speed, which affects mechanical friction.

Figure 4 shows the SFC sensitivity to the stroke-bore ratio range of 1.0 to 1.25. The change in simulated SFC over this range of stroke-bore ratio was only 0.4 g/kW-hr. The cycle simulation used to evaluate SFC changes did not include crevice volume effects or the interaction between combustion chamber shape and combustion rate. The simulation did include changes in heat transfer surface area, valve sizes, and mechanical friction.

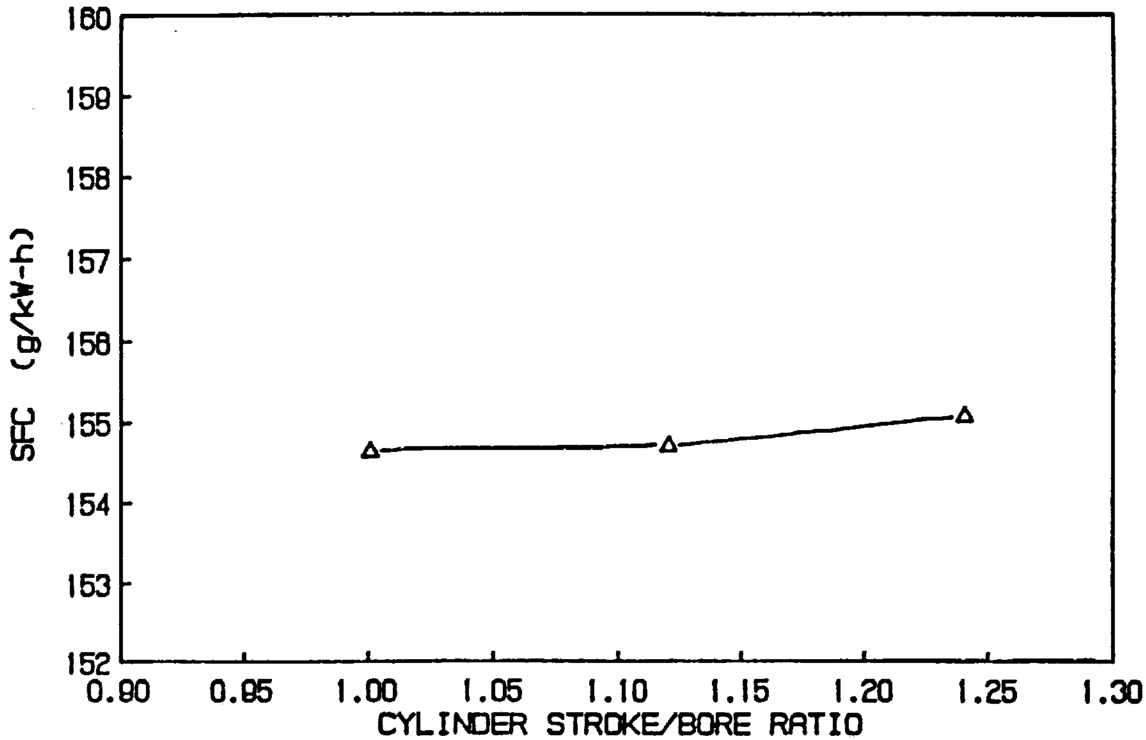


Figure 4 - SFC Sensitivity to Stroke/Bore Ratio

SFC Sensitivity to Heat Transfer

The change in SFC due to reduced heat rejection from piston, cylinder head, cylinder liner, and exhaust port and manifold were investigated using the engine cycle simulation program. The predicted change in SFC due to piston heat rejection is shown in Figure 5.

A heat rejection level of 3700 Watts is predicted for a ferrous piston cap with under crown oil cooling. An articulated piston with a ferrous crown and aluminum skirt was assumed for all the engine analyses due to structural requirements for the high peak cylinder pressures. The change in SFC expected by reducing the piston heat rejection from 3370 W to 900 W is 2.4 g/kW-hr. The potential 1.5% SFC reduction due to reduced piston heat transfer makes the piston a prime candidate for advanced insulation options.

Various methods of insulating the cylinder head bottom deck were investigated during the component concept task. The reduction in SFC predicted for cylinder head insulation is shown in Figure 6.

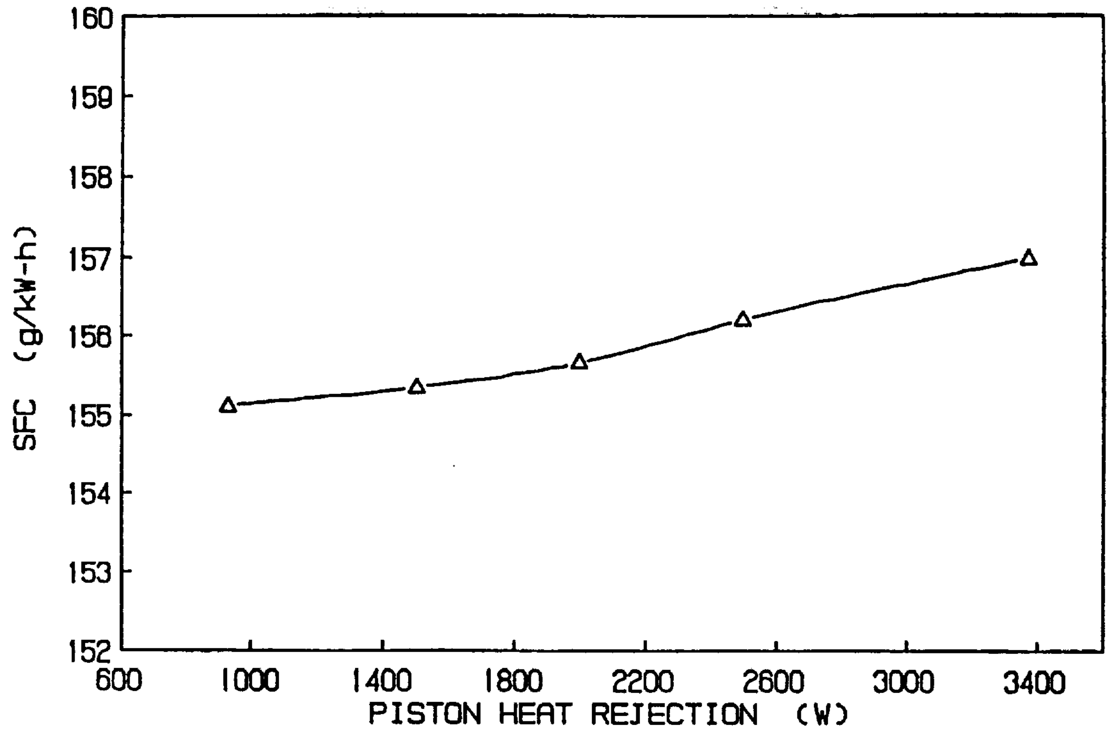


Figure 5 - SFC Sensitivity to Piston Heat Rejection

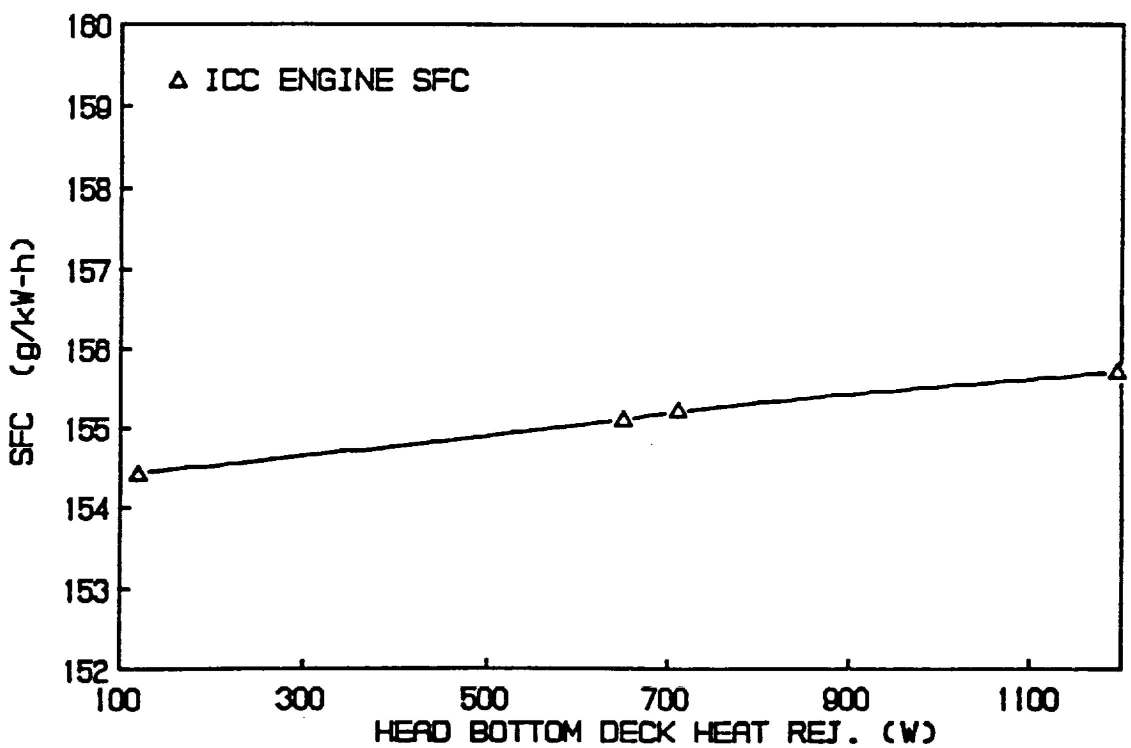


Figure 6 - SFC Sensitivity to Cylinder Head Heat Rejection

The maximum cylinder head heat rejection of 1200 Watts shown in Figure 6 is for a metal, oil cooled bottom deck. The two points near 700 Watts heat rejection represent low conductivity ceramic bottom deck inserts. The minimum value of cylinder head heat rejection assumes an ideal air gap of 3 mm, which cannot be achieved in practice, but illustrates the SFC opportunity. The total SFC reduction potential, with the insulated cylinder head bottom deck, is 1.3 g/kW-hr.

The concept engine configuration has a turbocompound stage, so a reduction in exhaust port heat rejection will reduce SFC due to the higher exhaust temperature. Figure 7 shows the predicted change in engine SFC with reduced exhaust port heat rejection. The SFC decreases by 1.1 g/kW-hr with a reduction in heat rejection from 1870 W to 510 W. The intermediate heat rejection level of 950 W is viewed as being feasible with current exhaust port air gap designs.

The exhaust manifold without insulation is expected to reject 5000 Watts of energy to the engine compartment. With insulation, a modest SFC reduction of 0.5 g/kW-h is expected, if the heat rejection is lowered to 700 Watts. Exhaust manifold insulation will reduce the surface temperature of the manifold to control the fire hazard from a fuel leak.

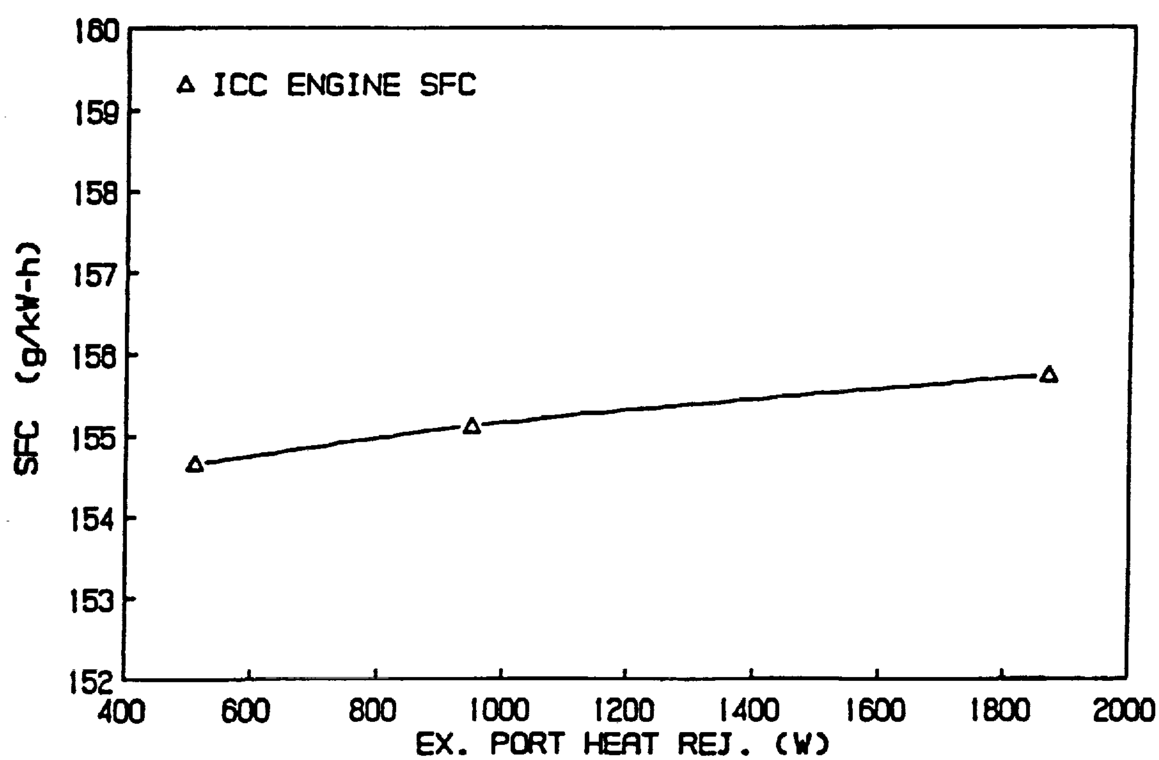


Figure 7 - SFC Sensitivity to Exhaust Port Insulation

A cast iron cylinder liner was evaluated with both oil cooled and uncooled designs. The most important effect of insulation on the cylinder liner is the change in the surface temperature near top ring reversal (TRR). The liner wall temperature plays a major role in determining the lubricant requirements and the wear life of the liner and piston rings. The cylinder liner material selection and cooling design is more dependent on the tribological system than the performance effects. Profiles of the calculated cylinder liner inner wall surface temperature are shown in Figure 8 for an oil cooled and uncooled design. Removal of the oil cooling increases the top ring reversal temperature from 229 to 382°C and decreases SFC from 155.1 to 154.5 g/kW-hr. The uncooled liner prediction included a small decrease in parasitic load to account for the lower oil pump power required.

Heat rejection levels of the in-cylinder components selected for evaluation of the baseline 22 MPa PCP engine system are shown in Table 4. The decrease in SFC for each component is compared to a conventional metal oil cooled design. The total SFC benefit of the insulation package shown in Table 4 is 4 g/kW-hr. The 22 MPa PCP baseline engine concept has a predicted SFC of 155.1 g/kW-hr.

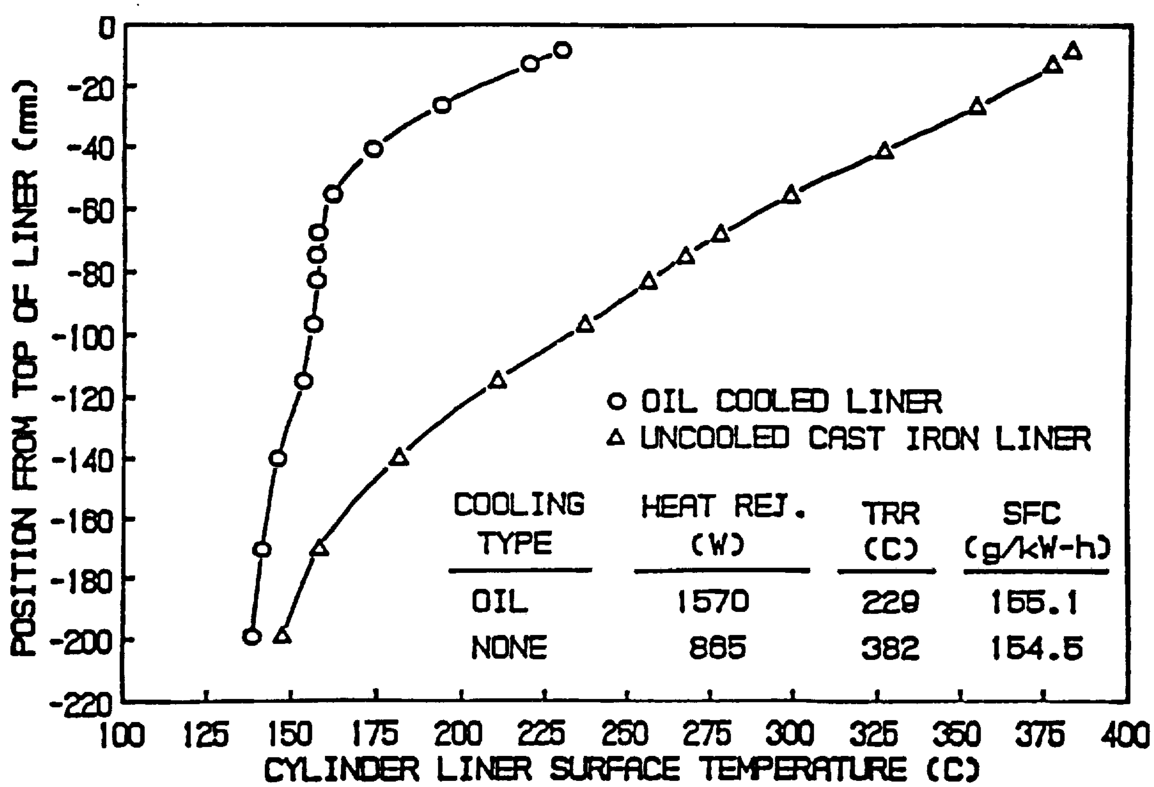


Figure 8 - Predicted Cylinder Liner Temperatures

Table 4 - In-Cylinder Component Heat Rejection Summary

COMPONENT	Heat Rejection Max(W)	Heat Rejection Min(W)	BSFC Change g/kW-hr
Piston	3369	926	-1.9
Cylinder Head Bottom Deck	1194	119	-1.3
Exhaust Port	1870	510	-1.1
Cylinder Liner	1570	865	-0.7
Exhaust Manifold	5014	718	-0.5

SFC Sensitivity to Air System Variables

One of the technology building blocks to achieve a 152 g/kW-h SFC is the use of increasing PCP with optimum BMEP. The BMEP for minimum SFC at a given cylinder pressure limit increases as cylinder pressure increases. The rated power, air-fuel ratio must remain relatively constant to obtain minimum SFC and particulate emissions. The boost pressure must increase with increasing BMEP. Simulation results have shown that compressor pressure ratio must increase linearly with increasing engine BMEP over a pressure ratio range from 2.7 to 6.1.

Turbomachinery efficiency has a major impact on overall engine efficiency as the turbocharger compression and expansion work increases and turbocompounding is added. The SFC sensitivity to single stage and series turbocharger efficiency is shown in Figure 9. The minimum turbocharger efficiency of 58% (total to total - compressor and turbine) is typical for production turbochargers supplied on current diesel engines. The single stage turbocharger, 19 MPa PCP engine SFC can be reduced from 167.6 to 158.6 g/kW-hr by increasing the turbocharger efficiency from 58 to 78%. A similar SFC sensitivity was found for the series turbocharged, 22 MPa PCP engine, with SFC decreasing from 163.1 to 153.7 g/kW-hr with the same efficiency change.

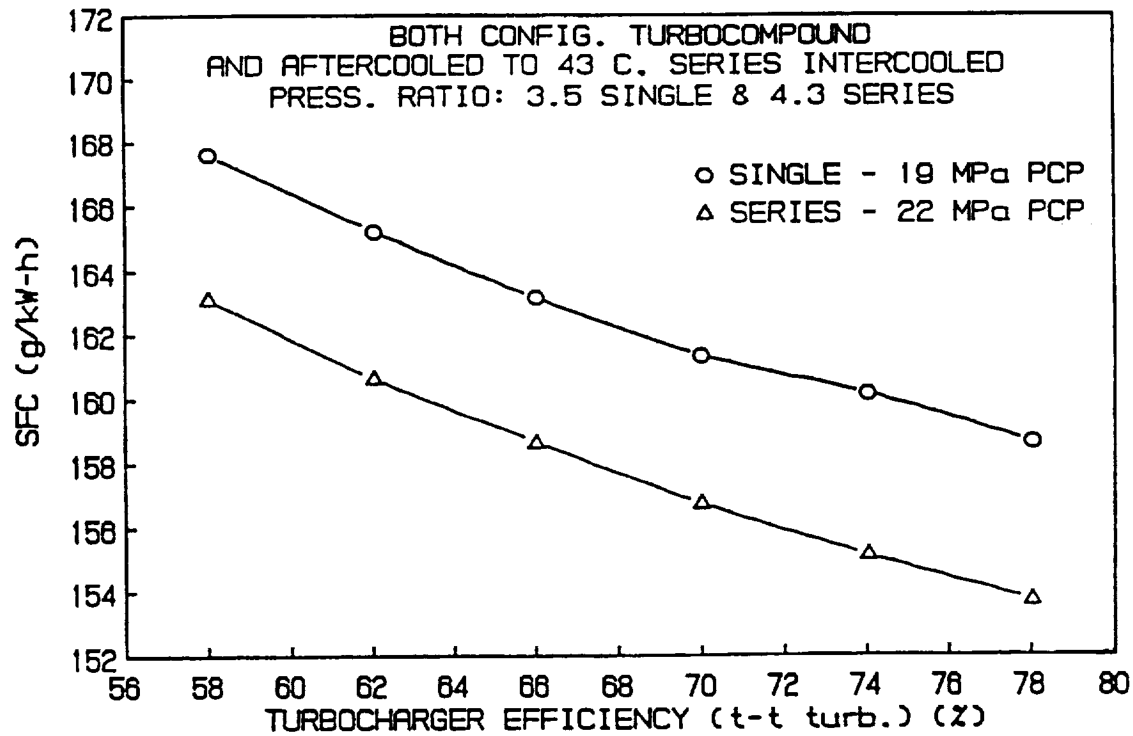


Figure 9 - SFC Sensitivity to Turbocharger Efficiency

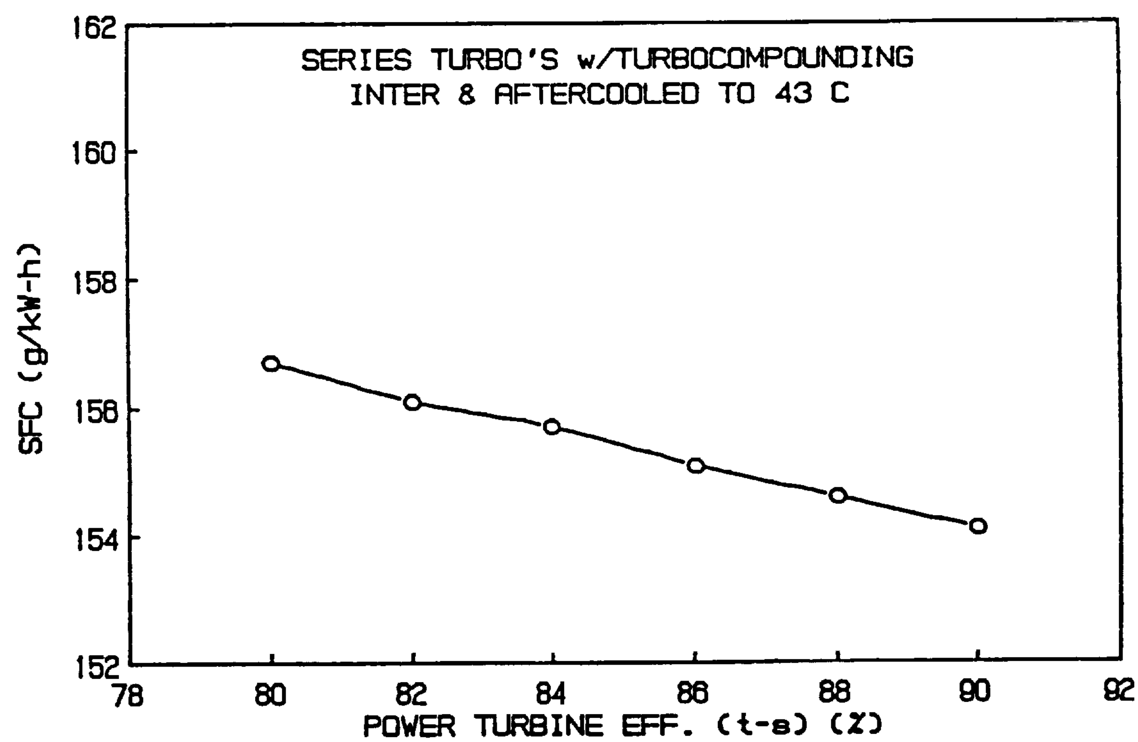


Figure 10 - SFC Sensitivity to Compound Power Turbine Efficiency

The change in engine SFC with turbocompound power turbine efficiency is shown in Figure 10. A 10% increase in turbine efficiency decreases SFC by 2.6 g/kW-hr. The minimum compound turbine efficiency shown is 80%. The total SFC reduction opportunity for increasing turbomachinery efficiency from current production levels, to advanced turbomachinery efficiency levels is on the order of 10 g/kW-hr. The concept engine systems in this study used turbomachinery efficiencies which represent the best aerodynamic technology currently available.

In the design of a turbocompounded engine, selection of the cross-sectional area in the power turbine can be used to determine the power turbine expansion ratio and the ratio of exhaust manifold to intake manifold pressure. The engine back pressure/boost ratio is a design variable and the relationship with rated SFC is shown in Figure 11. Power turbine expansion ratio increases with back pressure/boost ratio. Rated power SFC is minimum near a value of 1.20. A value of 1.25 was selected for the 22 MPa baseline engine.

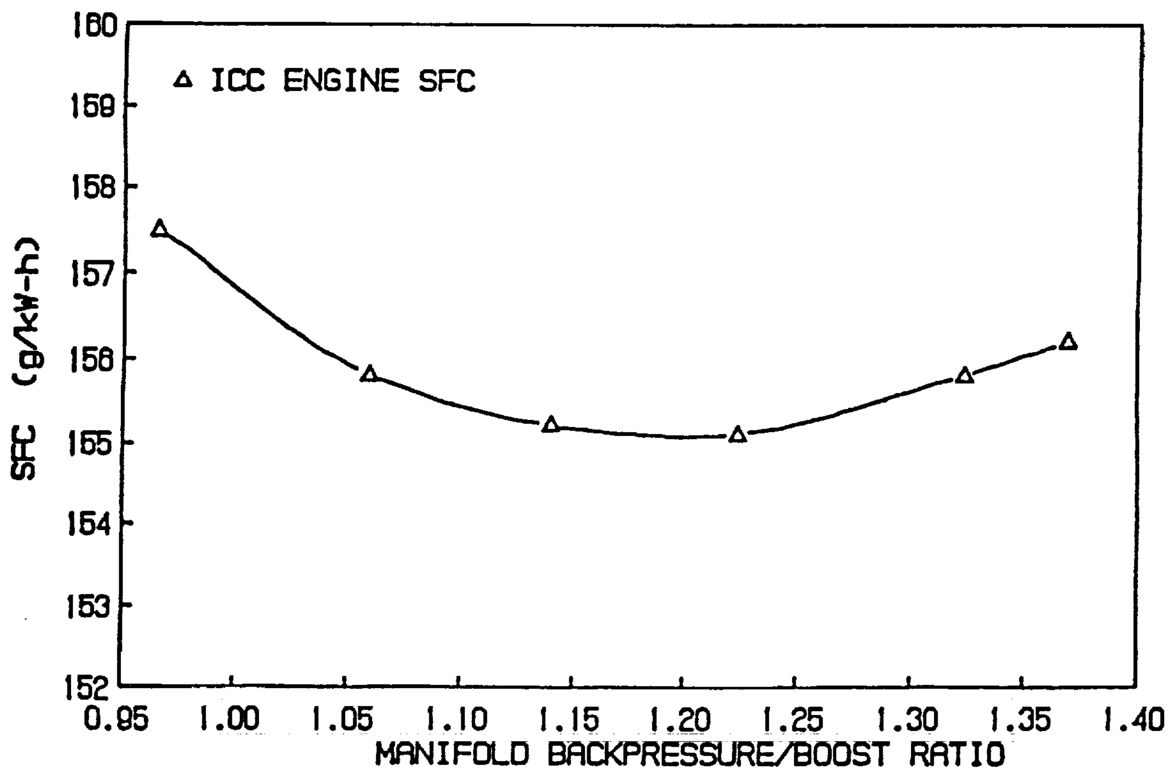


Figure 11 - SFC Sensitivity to Backpressure/Boost Ratio

Intercooling and Aftercooling

Intercooling or aftercooling of the inlet air are engine design options. Air-to-air heat exchangers for intercooling and aftercooling to 43°C were assumed for the baseline 22 MPa PCP concept engine. If the intercooling between the low and high pressure compressor stages is removed, SFC increases by 2.4 g/kW-hr.

The engine cycle simulation results indicate that SFC does not increase when aftercooling is removed. If exhaust energy were used to increase the intake temperature to 300°C, the SFC would decrease by 2.2 g/kW-hr, but NOx emissions would be expected to increase significantly. Sub-ambient aftercooling to 0°C provides a small decrease in specific fuel consumption, but may prove to be more important as a strategy to reduce NOx emissions to levels below 5 g/hp-hr.

The engine cycle simulation was used to determine the impact of intercooling inlet air to 0°C. The 22 MPa PCP engine SFC would be reduced by an additional 2.3 g/kW-hr if sub-ambient intercooling could be obtained without any increase in parasitic loss to the engine.

A wide range of intake manifold air temperatures (0 - 300°C) was examined to define the SFC sensitivity to charge air temperature. The results of the inlet air temperature affect on SFC for the base line engine are shown in Figure 12.

Concept Engine Friction

The friction (FMEP) level of the 7.0 liter concept engine, at 1600 rpm and 22 MPa PCP, has been estimated to be 126 kPa. This friction estimate is 4.5% of the engine brake power at rated conditions. Increasing engine BMEP is one method of decreasing the relative size of friction losses. Concept engine cylinder friction losses were lowered by limiting the average piston speed to 6.6 m/s. Figure 13 shows the sources of the concept engine friction and their effect on SFC. If engine friction could be eliminated, the SFC would decrease by only 5.5 g/kW-hr.

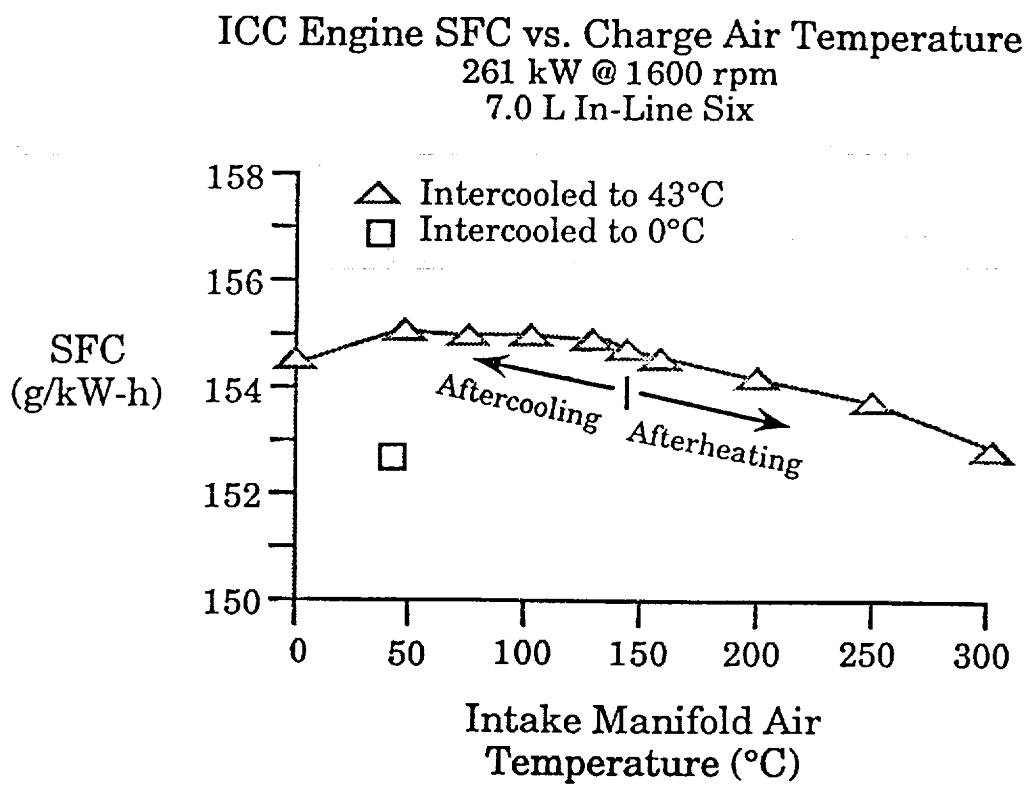


Figure 12 - SFC Sensitivity to Charge Air Temperature

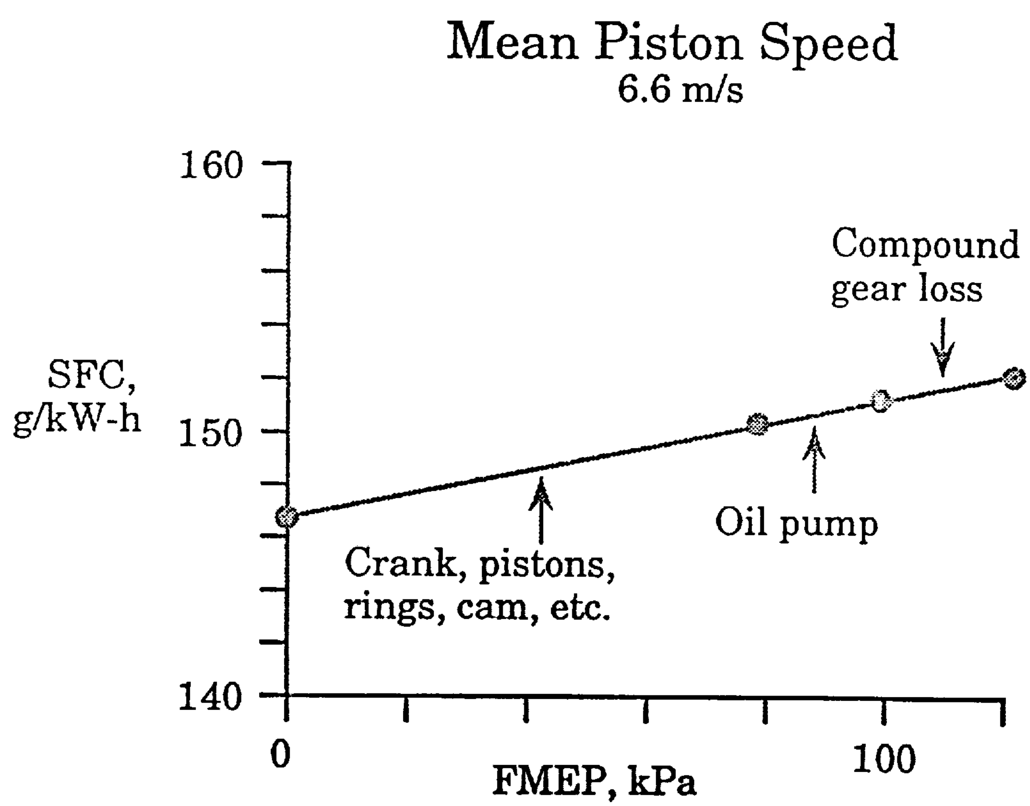


Figure 13 - SFC Sensitivity to Friction Losses

Analysis of a Concept Rankine Bottoming Cycle System

A single stage turbocharged engine with 19 MPa cylinder pressure limit and turbocompounding has a predicted rated power SFC of 161.3 g/kW-hr. This engine configuration requires the addition of a bottoming cycle to approach the 152 g/kW-hr SFC goal.

A steam Rankine bottoming cycle (RBC) system was concepted and analyzed based on experience gained from Caterpillar's previous RBC development [1]. The initial assumptions for the 19 MPa PCP engine RBC design are given in Table 5.

TABLE 5 - BOTTOMING CYCLE ASSUMPTIONS

500 psi (3450 kPa) Maximum System Pressure

Expansion to 30 psi (207 kPa)

Condense at 250^{o}F (121^{o}C)

Preheat and Boiling in the Evaporator

70% Turbine and Feed Pump Efficiency

A single pressure system was selected to reduce the RBC system cost and complexity. A sample water temperature-entropy diagram for the steam bottoming cycle system is shown in Figure 14. The initial cycle included superheat from 467^{o}F to 710^{o}F at 500 psi and then expansion to 30 psi. This 19 MPa PCP engine system cycle produced a reduction in SFC from 161.3 to 155.9 g/kW-hr.

Various combinations of turbine inlet pressure and condenser pressure were evaluated to improve the system performance. The analysis results are shown in Table 6. The expansion ratio was held constant at 40:1 as turbine inlet pressure and condenser pressures were decreased. With lower turbine inlet pressure, the saturation temperature is lower, which increases the amount of energy extracted from the exhaust stream. The total engine SFC decreased to 153.4 g/kW-hr.

TABLE 6 - RBC OPERATIONAL PARAMETERS

Turbine Inlet Pres. psia	100	150	200	300	400
Condensing Pressure psia	2.5	4.0	5.0	7.5	10.0
BSFC g/kW-h	153.4	153.7	153.9	154.4	154.7
Vaporizer					
Heat Recovery kW	95.8	89.7	85.3	78.2	72.6
Area sq. ft.	336	326	318	304	291
Volume cu. ft.	5.2	5.0	4.9	4.7	4.5
Condenser					
Heat Rejection kW	79.4	74.4	70.5	64.5	59.8
Temperature °F	134	153	163	180	193
Area sq. ft.	6.2	4.4	3.7	2.8	2.3

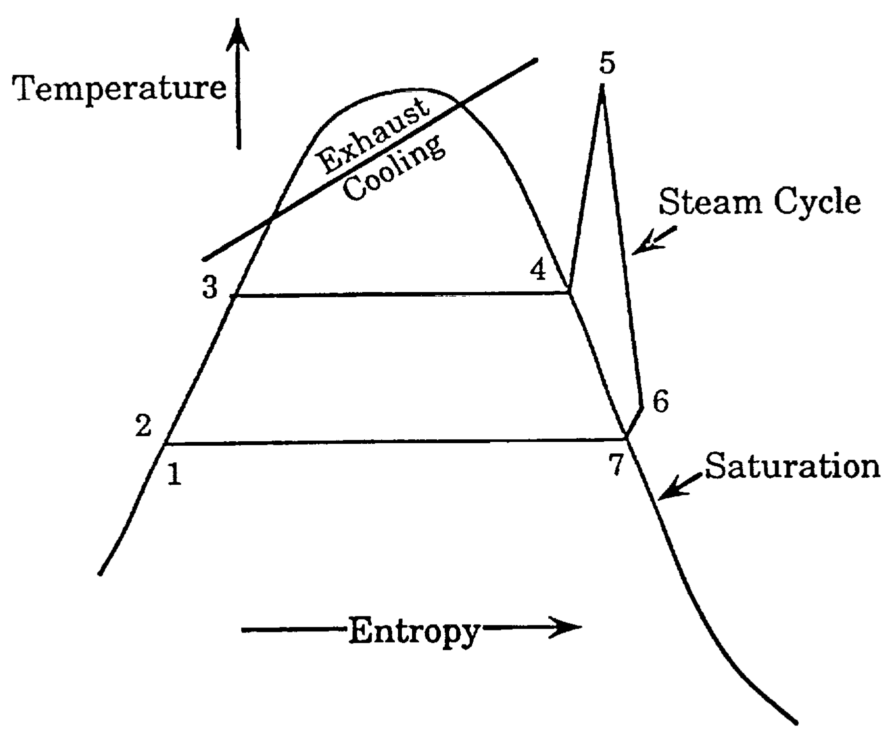


Figure 14 - H-S Diagram for Rankine Bottoming Cycle

As the RBC operating pressures decreased, the estimated sizes of the vaporizer and condenser increased. The vaporizer increases in size because of the increase in recoverable energy from the exhaust stream. The condenser gets larger due to a decrease in the saturation temperature and an increase in the amount of heat to reject. Total volume for the heat exchangers given in Table 6 increases by 68% for the range of pressures shown.

The RBC system with a maximum pressure of 150 psi (1034 kPa), a condensing pressure of 4 psi (28 kPa), and a rated SFC of 153.7 g/kW-hr was selected for further analysis. A schematic drawing of the 19 MPa PCP concept engine with bottoming cycle is shown in Figure 15. The 19 MPa PCP, 10 liter concept engine incorporates a single stage turbocharger, turbo-compounding, and bottoming cycle to yield a base engine power of 261 kW.

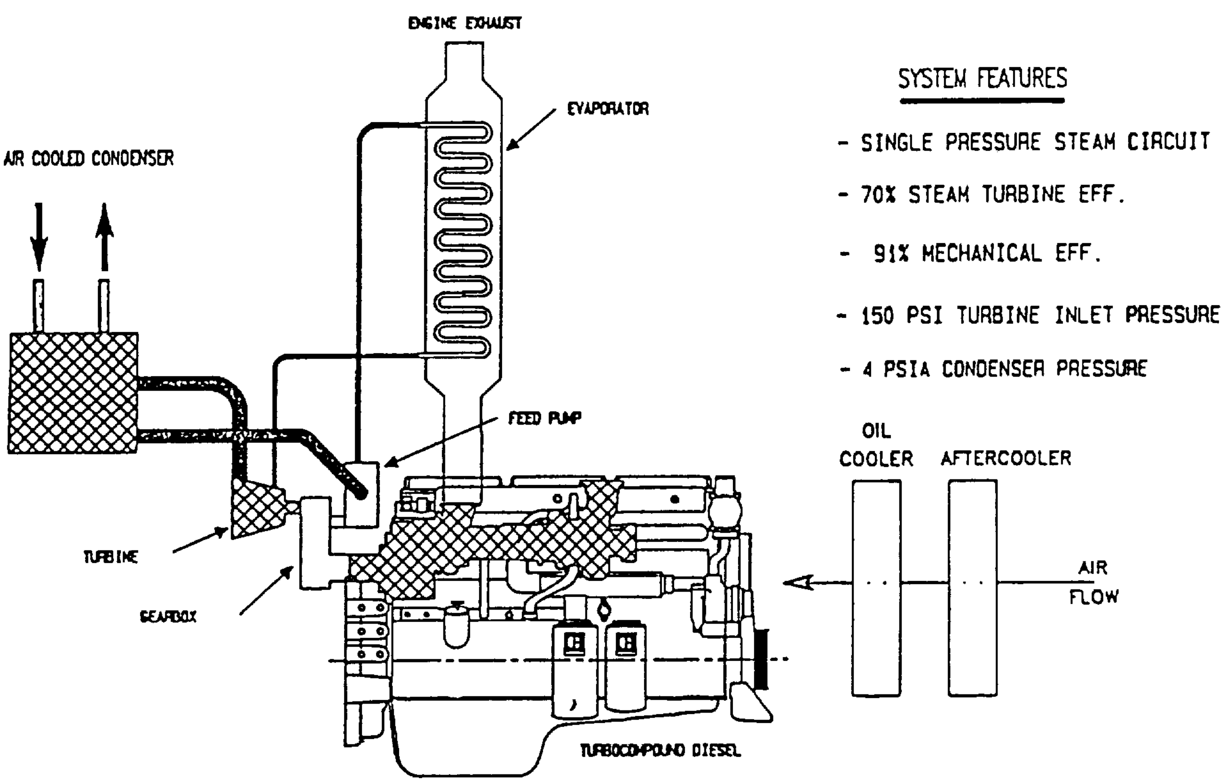


Figure 15 - 10 L Engine with Rankine Bottoming Cycle

Description of Candidate Engine System Concepts

The engine system concepts were constructed from the results generated in the concept analysis. The engine PCP was a basic design variable used to differentiate the system concepts. A description of the three engine options is given in Table 7.

The 19 MPa PCP concept engine required a bottoming cycle system to approach the 152 g/kW-hr SFC goal. Additional simulation work with turbomachinery maps reduced the predicted rated power SFC for the 19 MPa PCP engine to 152.7 g/kW-hr. The 19 MPa PCP engine concept has a single stage turbocharger and 10.3 liters displacement. All three engine concepts have low heat rejection in-cylinder components and a turbocompound system.

A 7 liter displacement was selected for both the 22 and 24 MPa PCP concept engines to take advantage of the higher optimum BMEP. The 22 and 24 MPa PCP concept engines utilize high efficiency, series turbochargers with intercooling, and a turbocompound stage. The 24 MPa PCP concept engine was formulated to meet the 152 g/kW-hr goal. The 24 MPa PCP engine cycle simulation changes made to lower SFC were a 0.5 increase in compression ratio, a 200 kPa increase in BMEP, and an increase in air/fuel ratio from 29 to 30. The three engine systems, summarized in Table 7, were used for the concept evaluation.

TABLE 7 - CONCEPT ENGINE SYSTEMS

SFC	152.7	155.0	152.1
Peak cylinder pressure, MPa	19	22	24
RBC	yes	no	no
Turbocompound	yes	yes	yes
Displacement, L	10.3	7	7
Rated speed, rpm	1600	1600	1600
BMEP, MPa (psi)	1.9 (275)	2.8 (406)	3.0 (435)
Turbochargers (pressure ratio)	single (3.03)	series (4.3)	series (4.7)
Turbocharger efficiency, %	69	74	76
Compression ratio	17	16	16.5
Insulation	Uncooled head with ceramic insert Oil cooled liner Low heat rejection piston		

III - IN-CYLINDER COMPONENTS CONCEPTS

In-cylinder component concepts were selected to meet the functional requirements of the candidate engine concepts. The in-cylinder component concepts were evaluated for their ability to meet the functional requirements and to satisfy the boundary conditions generated with the engine cycle simulation program. The potential for meeting the structural loads imposed by a 24 MPa PCP was one of the principal considerations. The in-cylinder components were also judged on their potential to meet the concept engine heat rejection goals shown in Table 8.

Table 8 - Component Heat Rejection Goals

Component	Heat Rejection - Watts
Piston	926
Cylinder Head Bottom Deck	650
Exhaust Port	950
Cylinder Liner	1570
Exhaust Manifold	718

The in-cylinder components were broken into the following categories for the initial concept work;

1) Pistons / insulation
2) Combustion chamber geometry
3) Cylinder head / insulation
4) Valve types / actuation
5) Bearing systems / connecting rod / crankshaft
6) Cylinder liner / cooling
7) Lubrication systems
8) Fuel injection concepts
9) Materials applications / insulation / wear

The more promising in-cylinder components from the concept phase will be reviewed in the following sections.

Piston Concepts

Piston concepts were first evaluated for their structural capability to meet the 24 MPa PCP goal. A two dimensional (2D) model was used during the concept work to evaluate the piston crown operating stresses and ring groove temperatures. Boundary conditions for the 2D finite element (FE) model were available from the concept engine cycle analyses.

All piston concepts were low heat rejection configurations because piston heat loss has a significant impact on engine SFC. The choice of the piston crown material, the insulation configuration, and crater geometry have a significant impact on the piston operating temperatures and thermally induced stresses.

Steel, ceramic, and aluminum crown materials were considered in the concept phase. Aluminum, and aluminum fiber reinforced derivatives, were considered for the piston crown, but were discarded due to the lack of strength at the proposed operating temperatures and pressures. Composite piston crowns, utilizing steel and ceramics, were considered viable options. Monolithic ceramic and ceramic coatings were crown material options.

Plasma sprayed ceramics (thermal barrier coatings) have shown potential for reducing heat loss through a piston. Problems in applying the coating system and the coating system stresses are strongly influenced by the piston crater geometry. A combustion chamber concept study was conducted to evaluate piston crater shapes that would minimize coating application and operating stress problems.

The piston structure transmits the cylinder pressure loads to the connecting rod through the piston pin. A number of piston pin configurations were evaluated in conjunction with the piston concepts. The piston pin has a rotary motion relative to the piston and connecting rod that requires a bushing and lubrication. Tribology options for the piston, rod and pin have to be considered in the evaluation of the piston concepts.

Piston ring materials and ring lubrication concepts are an important factor in evaluating low heat rejection piston concepts. Piston ring and groove temperatures must be selected based on the piston ring materials strength and the deposit forming properties of the lubricant. High piston ring groove temperatures will promote deposits that interfere with normal ring movement and lead to ring scuffing and breakage.

Piston Concept Selection

The piston selection required trade-offs between structural integrity, combustion chamber (including piston crater geometry), materials, insulation, and the piston pin design. Four piston pin options, shown in Figure 16, were included in the piston concept evaluation.

Antifriction bearings were considered during the concept phase as a method to reduce friction and SFC. Figure 16A is a piston pin concept incorporating an antifriction bearing shown in the size required for the predicted loads. This pin/bearing option could be used with either a conventional one piece piston or an articulated piston design. The antifriction bearing approach was not selected because the reduction in friction in the piston/pin joint was not large enough to justify the added cost of the antifriction bearing. There were also concerns about the long term durability of the rolling elements in this application due to the limited rotation of the piston/pin joint.

Figure 16B is a spherical piston/pin joint concept. The primary attraction for the spherical joint is the load carrying potential. The spherical joint design appeared to be limited to one piece piston bodies that had suitable bearing properties. The use of aluminum or fiber reinforced aluminum has been ruled out as a piston crown material due to the predicted loads and temperatures. Applying a bearing material to the socket in a steel piston appeared to have significant cost and manufacturing penalties. Piston socket deformation to achieve the load carrying capacity was an item of concern. A finite element model of the spherical piston/ rod joint was not constructed to evaluate the piston deflections and stress levels.

Figure 16C is the cross section of an articulated piston geometry that uses a separate piston crown and skirt that are tied together by the piston pin. This configuration permits the use of different materials selected to match the operating stresses. The piston crown was analyzed using several different steel materials in combination with thermal barrier coating and air gap insulation options. The piston skirt could be made from either an aluminum or iron material that provides a suitable bearing surface with the cylinder liner. Aluminum is the first choice for a skirt material for cost, weight and bearing considerations. Piston skirt structural (deflection) considerations will be a limiting factor in selecting the skirt material.

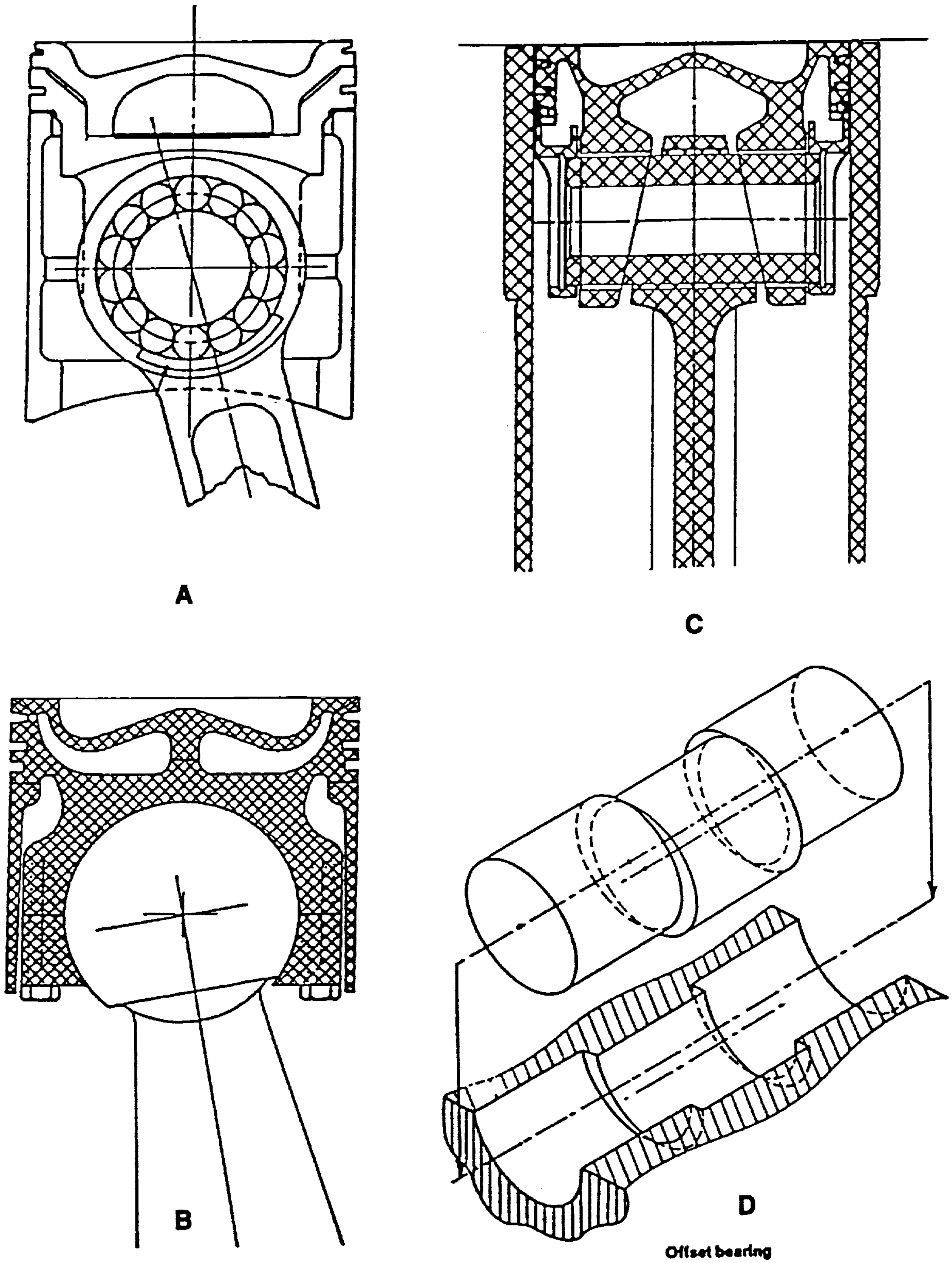


Figure 16 - Piston Wrist Pin Concepts

Figure 16D is a variation of a cylindrical piston pin. This pin configuration creates a piston pin rocking motion that provides positive lubrication for the piston pin bushing under high cylinder pressure operation. The offset piston pin geometry is a more expensive configuration to produce. Piston pin bushing analysis indicated that a cylindrical pin would be acceptable, if pressure lubrication was supplied to the bushing through a drilled passage in the connecting rod.

The articulated piston configuration was selected for further concept evaluation because it had the greatest flexibility for meeting insulation and structural requirements. The piston illustrated in Figure 17 is a sketch of a bolted and articulated piston design. This configuration permits maximum flexibility in selecting material to meet the operating temperatures and stresses. For example, the piston crater could be made from a monolithic ceramic to meet the insulation and high temperature requirements. The piston ring belt carrier could be made from a less expensive steel to provide the necessary wear resistance in the piston ring grooves. The lower structure could be made from a low cost nodular iron that would have suitable bearing properties for the piston pin connection. The composite piston structure would be bolted together.

Piston insulation options analyzed included thermal barrier coating systems, air gap configurations and air gap with thermal barrier coatings. Figure 18 is a sketch of the articulated piston crown with a plasma sprayed zirconia, thermal barrier coating system. A thermal barrier coating system, in the range of 2.5 to 3.5 mm thickness, is needed to meet the target heat rejection goals. The thermal barrier system is similar to the system being tested in DOE/NASA contract DEN3-332.

Figure 19 is a concept drawing of a piston with a welded air gap crown. A welded piston crown was expected to be more cost effective than a bolted piston crown. The piston crowns in Figure 18 and 19 were analyzed using 2D finite element models to predict piston crown heat rejection and stress states. Several material options, SAE 8645, Waspaloy and 17-4 PH stainless steel, and thermal barrier coatings were analyzed for the air gap piston. Table 9 summarizes the heat rejection, maximum crown temperature, maximum metal temperature, and top ring groove (TRG) temperature for the insulated piston options.

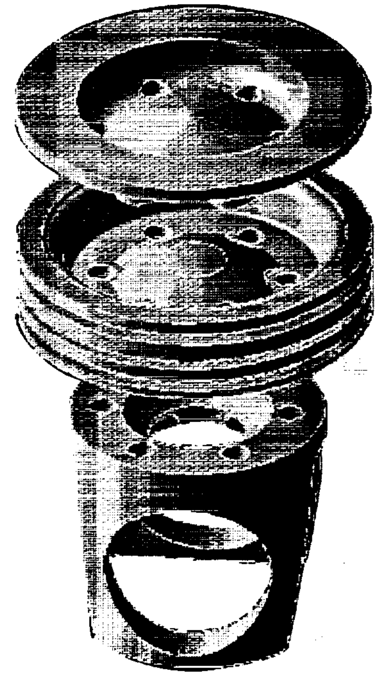


Figure 17 - Concept of Bolted, Articulated Piston

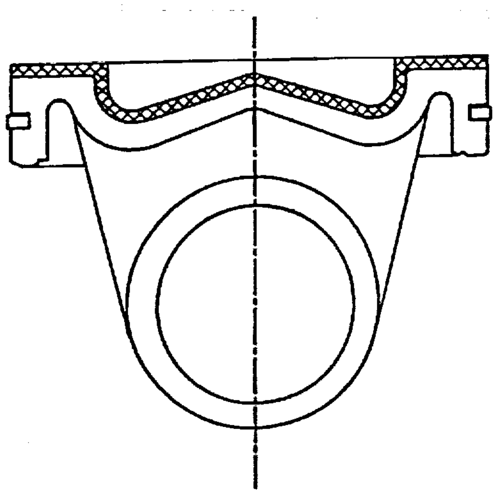


Figure 18 - Piston Crown with Thick TBC System

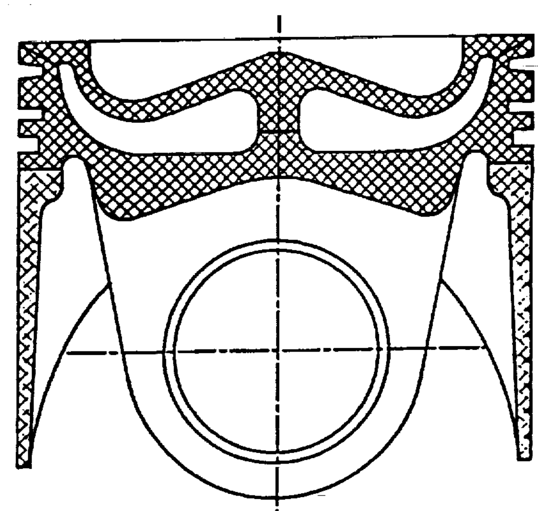
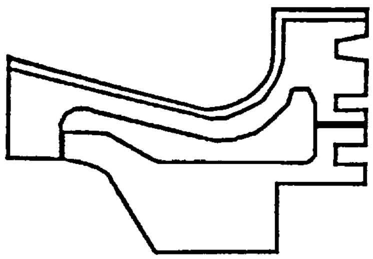


Figure 19 - Welded Air Gap Piston Concept

Table 9 - Insulated Piston Summary



Welded Air Gap TBC Piston
Heat Transfer Analysis Summary

	Q (Watts)	Max. (°C)	Max. Metal (°C)	Max. TRG (°C)
1.5 mm graded TBC/ 8645 steel crown/body	-1323	690	496	334
1.5 mm graded TBC/ Waspaloy crown/ 17-4 PH S.S. body	-974	701	600	345
No TBC/Waspaloy Crown/17-4 PH S.S. body	-1440	644	644	411
TBC (no air gap)	-929	716	313	278

The 3.5 mm thick thermal barrier coated piston without an air gap had the lowest predicted heat transfer and top ring groove temperature. Top ring groove temperatures must be limited to avoid deposits that can cause ring sticking. A top ring groove temperature limit of 380°C was selected based on Caterpillar experience with a high temperature synthetic lubricant. The Heavy Duty Transport program does have lubricant research work in progress that may allow the 380°C ring groove temperature to be increased.

A Waspaloy crown, with a 1.5 mm thermal barrier coating, welded to a 17-4 PH stainless steel body also met the heat transfer criteria. The Waspaloy crown without the TBC system had unacceptable top ring groove temperature and heat rejection. Thermal barrier coated pistons as well as air gap pistons options were included in the Task 2 concept selection.

Combustion Chamber Concepts

A diesel engine combustion chamber is formed by the cylinder head bottom deck, the cylinder liner walls, and the top of the piston. Several combustion chamber options were evaluated during the component concept phase. The combustion chamber shape will have a strong influence on the design of the cylinder head and piston. Figure 20 is one of the combustion chamber options considered in the concept phase. The cylinder head bottom deck is flat for ease of manufacturing. The crater in the top of the piston forms the combustion space as the piston approaches top dead center. The deep piston crater can be economically produced in conventional metal pistons, but creates design and manufacturing challenges for low heat rejection piston designs.

A second combustion chamber geometry evaluated in the component concept phase is illustrated in Figure 21. This configuration incorporates a contoured cylinder head bottom deck and a smaller crater in the piston. This configuration places more of the combustion space in the head. This provides greater flexibility for the design of low heat rejection pistons at the expense of a more complicated head geometry. The piston height is reduced with this configuration, which is an added benefit.

The contoured bottom deck of the cylinder head provides the option for more valve surface area, which is beneficial for improved air flow. Increased valve port area was evaluated as part of the cylinder head concept work. The contoured cylinder head bottom deck would increase the cost of the cylinder head if the hemispherical surface had to be machined. Cylinder head design and machining options were evaluated during the Task 2 component economic analyses.

Cylinder Head Concepts

The cylinder head concepts had a number of competing functional requirements to be satisfied:

1) Structural life >10,000 hours at 24 MPa cylinder pressure,
2) Bottom deck high temperature strength and insulation,
3) Minimum or no head cooling,
4) Maximum valve and port flow areas,
5) Cost effective materials.

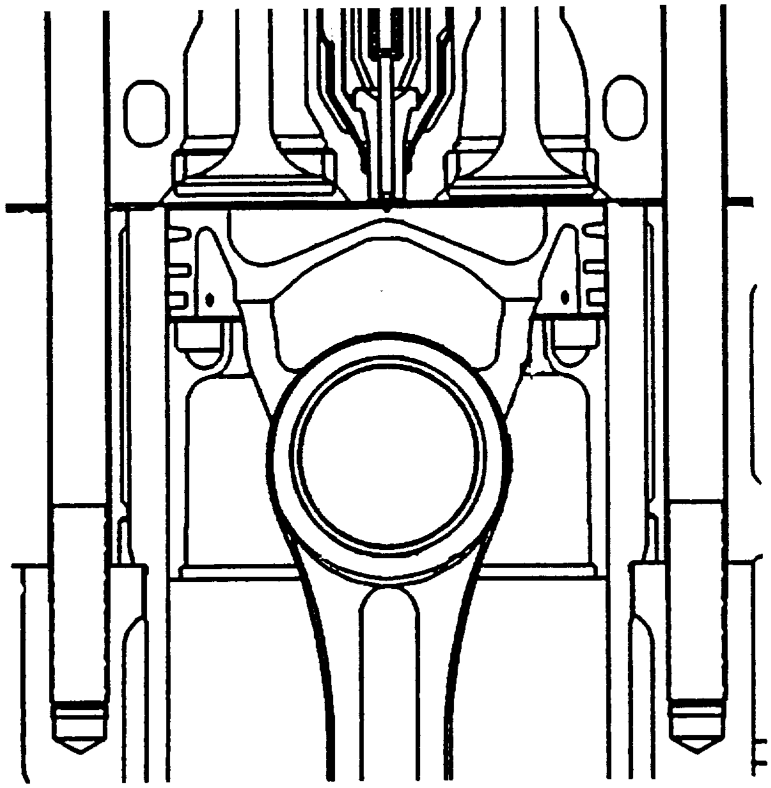


Figure 20 - Conventional Combustion Chamber Shape

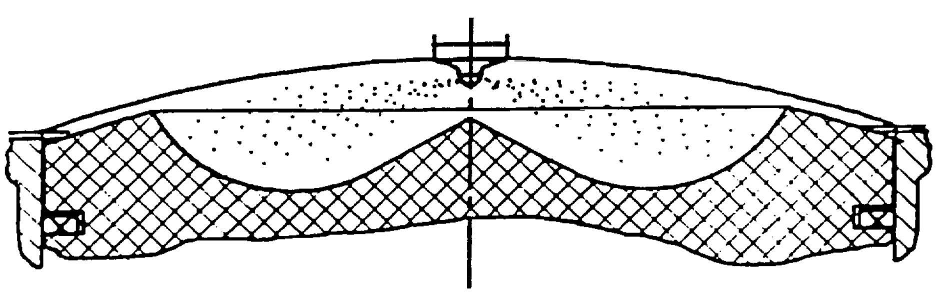


Figure 21 - Alternative Combustion Chamber Geometry

The cylinder head concept approach was to incorporate a high temperature bottom deck insert while utilizing current cylinder head casting technology and cast iron material for the rest of the head. The cylinder head insert concept is illustrated in Figure 22. The insert provides a number of functions:

1) Insulation for the cylinder head,
2) High temperature structural strength,
3) Valve seating surfaces,
4) Sealing surface for the cylinder liner,
5) Fuel injector location and insulation.

Monolithic ceramic and high temperature steel alloys were material candidates for the cylinder head insert. Materials were selected to meet the load and impact requirements for the valve seating surfaces and the cylinder liner-insert interface. Four head insert materials selected for evaluation in Task 2 included monolithic zirconia, whisker reinforced calcium alumina silicate (WRCAS), titanium and a 17-4 PH stainless steel. These materials were evaluated using a three dimensional (3D) finite element model of the cylinder head. Predicted heat transfer and probability of failure (POF) for the ceramics or fatigue factor for the metals is summarized in Table 10.

Table 10 - Cylinder Head Insert Evaluation

Material Options	Q-Watts	POF or FF
Zirconia (monolithic)	634	1.9×10^{-5}
WRCAS*	834	0.048×10^{-5}
Titanium	1374	FF > 2.0
17-4 PH (SS)	1554	FF > 2.0

*Whisker reinforced calcium alumina silicate

Life goal - POF $< 2 \times 10^{-5}$ or FF > 2.0

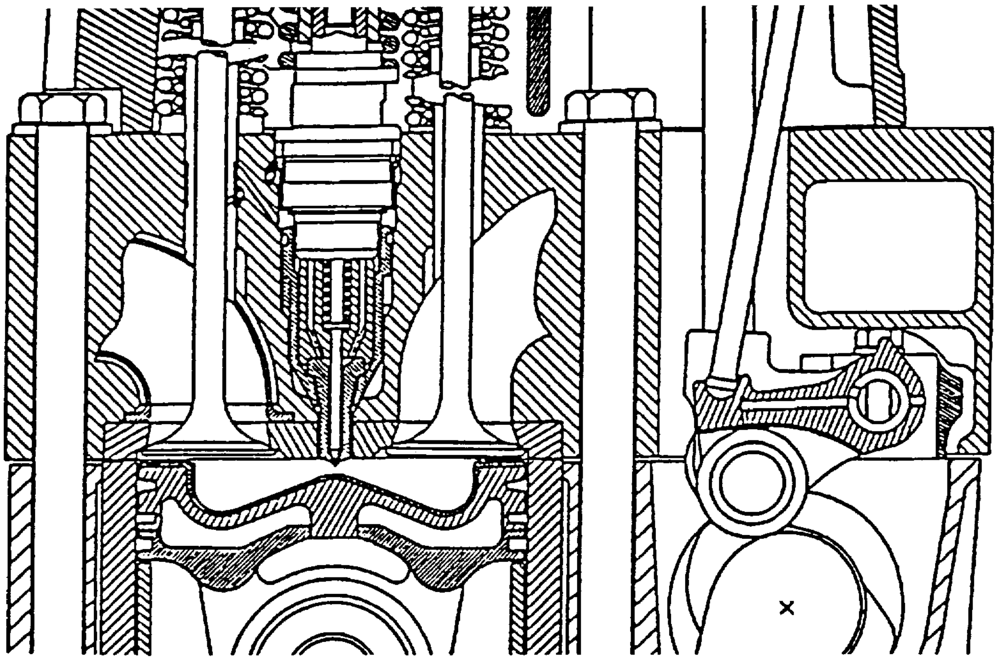


Figure 22 - Cylinder Head Cross Section

Intake and exhaust ports are integrally cast into the cylinder head. Preliminary analyses indicated that the exhaust port must be insulated for both structural and performance reasons. The heat loss to the air in the intake manifold was not considered to be large enough to warrant intake port insulation. Cast-in-place ceramic insulation for the exhaust port was considered but rejected in favor of air gap insulation. Ceramic insulation requires much greater space to achieve the same level of insulation. The air gap insulation scheme is illustrated in Figure 22. Both ceramic and metal port liners that form the gas passage were considered to be cost effective options.

The fuel injector location is shown in Figure 22. An unit injector fuel system was selected because the system has high pressure fuel delivery capability and can be electronically controlled. The operating temperature of the unit injector is controlled by fuel cooling. The initial cylinder head cooling concept was to use minimum cooling in the cylinder head to minimize in-cylinder heat loss. Selective oil cooling of the unit injector and the valve guides may be required. The cylinder head cooling options will be explored as part of the Phase II bench test evaluation. Limited cylinder head top deck cooling results from the lubrication of the valve train.

Valve and Valve Train Concepts

Several novel valve concepts, such as rotary valve porting, were considered during the concept phase, but were rejected in favor of a more conventional valve configuration. Structural integrity and valve sealing at 24 MPa cylinder pressure were the overriding considerations.

A 2D FE analysis was completed to evaluate valve structural life at the higher cylinder pressures and temperatures. The results of the FE analysis for an exhaust valve is shown in Figure 23. High cycle fatigue life and valve yielding will be acceptable if a Pyromet 31 material is used for the valve head. The Pyromet 31 can be welded to a 4140 stem material.

Several valve train concepts were evaluated in Task 1. The hemispherical combustion chamber option, shown in Figure 21, had the potential for increased valve area, but required design compromises due to the valve orientation. The valve and head insert for this combustion chamber geometry is illustrated in Figure 24. A mechanical valve train for this configuration would be more expensive. The hemispherical combustion chamber valve train would be a candidate for an electro-hydraulic valve actuation system such as the one being developed in contract DEN3-329.

Electronic control of the valve events provides the option for variable valve timing. Variable valve timing and larger port areas were explored in some detail using the engine cycle simulation program. No significant improvements in steady state rated or part load SFC were identified by varying the valve events. The use of variable valve timing may have potential for reducing engine gaseous emissions during transient engine operation. This feature was not explored due to the time required to run engine cycle simulation for the EPA transient conditions.

The combustion chamber and valve train configuration illustrated in Figure 22 was selected for further evaluation in Task 2. The valve train and unit injector are mechanically actuated. Fuel injection was assumed to be electronically controlled. Selective oil cooling of the valve guides and fuel cooling of the unit injector were assumed in the Task 2 engine concept preliminary design and cost analysis.

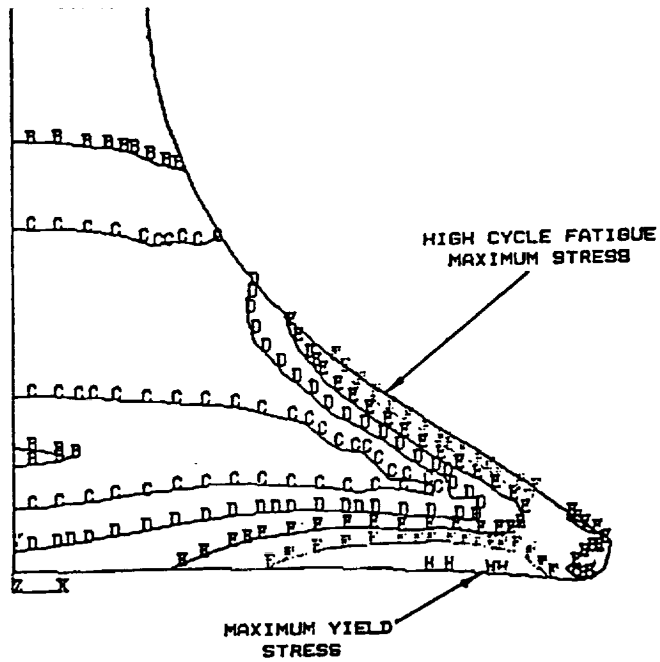


Figure 23 - 24 MPa PCP Exhaust Valve Stress Results

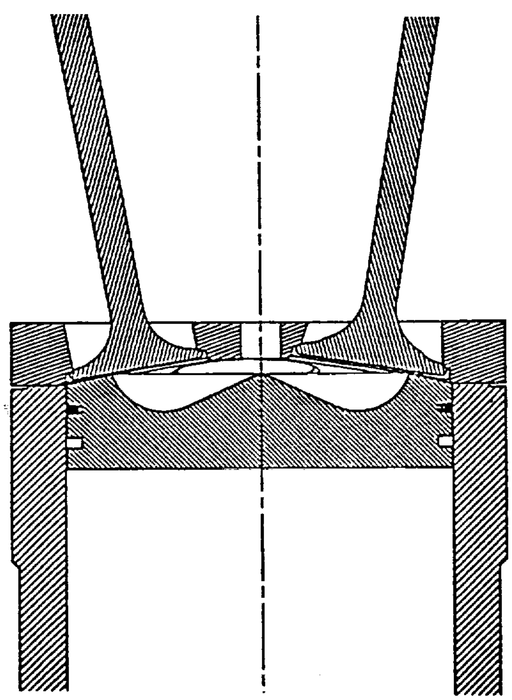


Figure 24 - Alternative Valve Configuration

Tribology Concepts

A number of lubricant options were explored during the Task 1 concept phase:

1) Vapor phase lubrication,
2) Solid lubricants,
3) Liquid lubricant with solid suspensions,
4) High temperature liquid lubricants.

The vapor phase lubrication, solid lubricants, and liquid lubricants with solid suspensions were evaluated for upper cylinder lubrication of the piston ring-liner interface. The primary concern was meeting the piston ring and liner durability goals at higher in-cylinder temperatures and pressures. Selection of bearing-lubricant options was treated as a second, but related lubricant system constraint.

Discussions were held with Dr. Elmer Klaus early in the concept work to review vapor phase lubrication options. The use of vapor phase lubrication for the upper cylinder, ring-liner area offered the potential for higher top ring reversal temperatures. The lubricant is delivered in a vapor phase through ports in the cylinder liner and converted to a solid lubricant on the hot ring and liner surfaces. Design problems associated with delivering controlled amounts of the lubricant to the top ring reversal area were considered to be beyond the time frame of the Phase 1 work plan. Vapor phase lubrication can result in excess wear if too little, or too much of the lubricant is supplied to the ring-liner interface.

The application of solid lubricants or a liquid lubricant with solid suspensions for the upper ring belt area was reviewed with Joe Cerini of General Technology. The use of solid lubricants for break-in appeared to be feasible. No practical method of continuously supplying the solid lubricant was identified.

Lubricating critical areas of a low heat rejection engine will require the use of a high temperature liquid lubricant. No other lubrication scheme is currently available that will meet the wear and durability goals for 500,000 miles between major overhauls. To achieve this life goal, the ring-liner pair must operate with a wear coefficient of 1X10E-9. The best known solid lubricant systems available today achieve wear coefficients on the order of 1X10E-6 to 1X10E-7, as illustrated in Figure 25. The solid film wear coefficient is two to three orders of magnitude short of the the desired wear goal.

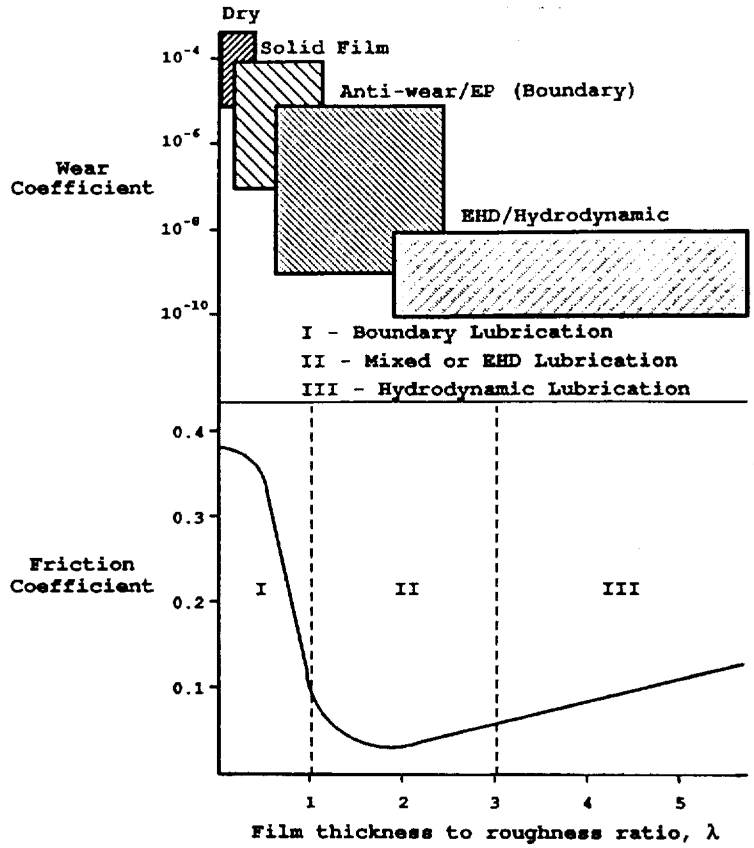


Figure 25 - Lubrication Regimes

The ring-liner tribological approach focused on a high temperature lubricant system to provides many of the same functions available in current diesel engines. The lubricant will need high temperature stability and viscosity. Piston rings, with plasma sprayed wear surfaces, will be required to meet the engine wear goals. Several ceramic and cermet piston ring wear face materials have been successfully applied to production piston rings. Ring face wear coatings will be tested in Phase II for compatibility with the cylinder liner material and high temperature lubricant options.

The cylinder liner concepts incorporated a liner bore wear surface for improved wear and scuff resistance. Several liner bore hard facing materials were included in the concept and analysis tasks. A ceramic chrome oxide wear surface is one liner option that has demonstrated good wear and scuff resistance. Several liner-ring material options are also being evaluated in a DOE/ORNL contract to develop wear resistant ceramic coatings for diesel engines.

Figure 25 indicates that the oil film thickness to surface roughness ratio is important in determining both wear and friction for the ring-liner combination. Oil film thickness is also a function of the operating conditions, piston-ring design and the viscosity of the lubricant. Liner surface finish and high temperature lubricant properties will be important factors in meeting the functional specification wear goals.

A three ring piston concept was evaluated using a piston ring simulation program available at C-K Engineering. The preliminary analysis indicated that the ring pack design would have acceptable ring dynamics and blowby at 22 MPa PCP. The ring pack model does not have the capability of predicting ring wear, but the piston ring forces and oil film thickness predictions were similar to current ring design practices.

Engine Bearing Concepts

The engine bearing and ring-liner lubrication requirements must be evaluated together, unless provision is made to provide separate lubrication systems. A common lubricant system, that provides component cooling, bearing lubrication and ring-liner lubrication, was assumed in the concept and analysis work. A single fluid lubrication system will be the most cost effective. Several high temperature lubricant candidates will be evaluated in the Phase II bench testing.

Typical engine bearing systems account for approximately one-third of the engine friction power consumption. A break down of the engine friction sources is shown in Figure 26. Piston ring and bearing friction account for one-half of the friction losses.

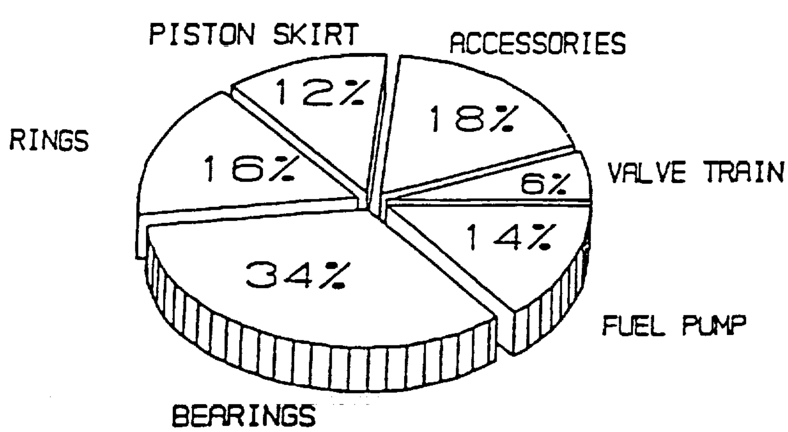


Figure 26 - Estimated Engine Friction Break Down

Rolling element and hydrodynamic sleeve bearings for the engine main bearings and connecting rod were evaluated in the concept phase. Rolling element bearings were sized for a B_{10} life of 10,000 hour and a shock factor of 2.0. The change in rolling element bearing width and power consumption relative to a hydrodynamic sleeve bearing for three candidate engine concepts is shown in Table 11.

Table 11 - Rolling Element Bearing Comparison to Sleeve Bearings

Bore (mm)	PCP (MPa)	Rolling Element Bearing Width Increase		Total Power Reduction	
		Rod	Main	%	Power (kW)
125	19	38%	0	30	2.0
110	22	38	0	23	1.3
110	24	37	0	26	1.7

A reduction in engine friction power loss (0.6 to 0.9 g/kW-hr reduction in SFC) is possible if sleeve bearings are replaced by rolling element bearings. On the negative side, the connecting rod bearings are significantly wider than comparable sleeve bearings with the same load capacity. Although rolling element main bearings will not require additional crank shaft length, wider connecting rod bearings will significantly increase the crankshaft and engine block length. The cost of a crankshaft incorporating the rolling element bearings would be higher than a conventional crankshaft with hydrodynamic sleeve bearings. For these reasons, rolling element bearings were not included in the engine concept design and cost analysis.

IV ENGINE PRELIMINARY DESIGN

In-cylinder and engine component concepts were evaluated for technical feasibility, reliability and durability potential, and potential impact on cost. The in-cylinder component contribution to reducing fuel consumption was given the highest priority. However, all in-cylinder component concepts were judged for the potential to meet the other engine functional specifications. The in-cylinder component concepts judged to have the potential for meeting the engine functional specifications were considered during the engine preliminary design.

In-Cylinder Component Engine Options

Three six-cylinder engine configurations used in the engine cycle simulation were included in the engine preliminary design work. The engine physical descriptions are listed in Table 12. Engine option 1 has a displacement of 10 liters with a 125 mm. bore and 140 mm. stroke. The engine is designed for a cylinder pressure of 19 MPa. The engine air system consists of a single stage turbocharger and a turbocompound unit. A Rankine bottoming cycle was added to the engine to approach the SFC consumption goal.

Engine options 2 and 3 have a displacement of 7 liters with a 110 mm. bore and 123 mm. stroke. The 7 liter engine will cover the desired power ratings at the higher cylinder pressures. Engine option 2 is designed for a cylinder pressure of 22 MPa. Engine option 3 is designed for 24 MPa cylinder pressure. Both engines use series turbochargers to achieve the BMEP ratings. The 7 liter engines incorporate a turbocompound stage to utilize the exhaust energy for reduced SFC and improved transient response.

TABLE 12 - IN-CYLINDER COMPONENT ENGINE OPTIONS

Option	1	2	3
Peak Cylinder Pressure - MPa	19	22	24
Displacement - Liters	10	7	7
Bore x Stroke	125 x 140	110 x 123	110 x 123
RPM	1600	1600	1600
Compression Ratio	17:1	16:1	16.5:1
BMEP - MPa	2.0	2.8	3.0
Turbocharger Configuration	single stage	series	series
Turbo Efficiency %	69	74	76
Turbocompound Stage	yes	yes	yes
Rankine Bottoming Cycle	yes	no	no
Predicted SFC *	154	155	152

*Injection timing for best SFC

Crankshaft - Block Preliminary Design

The engine block length for the three concept engines is determined by the structural requirements for the crankshaft and bearings. Hydrodynamic sleeve bearings were selected for the connecting rods and main bearings. Bearings were designed to meet the life criteria used for production engines. The crankshafts were designed to meet bending and torsional stiffness criteria.

The concept engines and in-cylinder components were compared to a Caterpillar production 3176 engine in the preliminary design and costing activities because the 3176 is approximately the same size. The size of the crankshaft for the 7 liter (110 mm. bore and 123 mm stroke) engine is shown in comparison with the 3176 engine crankshaft in Figure 27.

The higher operating cylinder pressures dictate larger bearing journals to meet bearing load carrying requirements. The crankshaft dimensions must also be increased to provide the necessary stiffness in bending at the higher cylinder pressures. The 19 MPa PCP,10 liter engine crankshaft is slightly longer than the 3176 engine crankshaft and has larger journal diameters. The 24 MPa PCP, 7 liter engine crankshaft is also the same length as the 10 liter engine crankshaft due the higher 24 MPa PCP bearing loads.

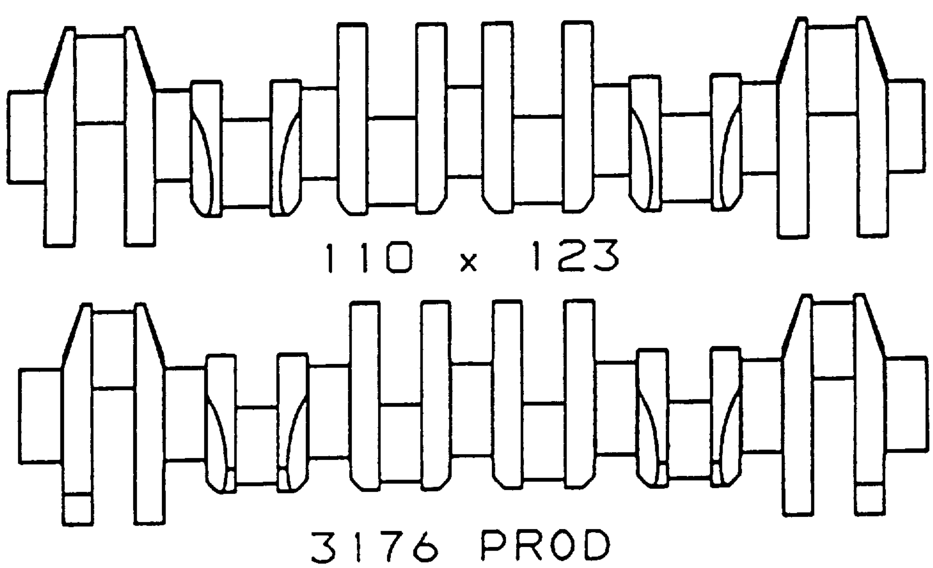


Figure 27 - Engine Crankshaft Size Comparison

The length and width of the engine block were determined by the crankshaft dimensions and structural requirements dictated by the cylinder pressure. The block height was determined from the piston and connecting rod dimensions. Figure 28 is a cross section of the concept engines showing the relationship.

The overall engine height, without turbomachinery, is determined from block, oil pan, cylinder head and valve train structural requirements. The oil pan shown in Figure 28 has the lubricant capacity for a 250 hour oil change interval. The lubricant sump capacity and oil pan dimensions may change, depending on the lubricant and lubricant change interval.

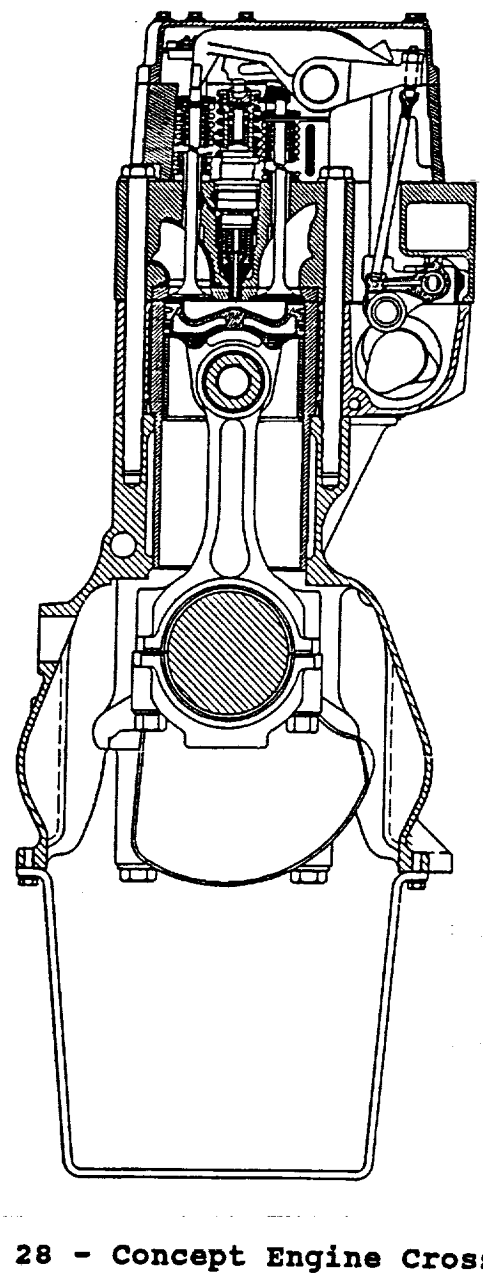


Figure 28 - Concept Engine Cross Section

Connecting Rod Preliminary Design

Larger connecting rod bearings and piston pins are required to meet the higher cylinder pressure loads for the 10 liter and 7 liter engines. A comparison between the 7 liter, 24 MPa PCP concept engine connecting rod and a production 3176 connecting rod is illustrated in Figure 29. The crankshaft bearing for the 7 liter engine connecting rod is wider and has a larger diameter compared to the production rod due to the higher cylinder pressure loads.

The bearing area for the piston pin has also been increased for the higher loads. The dashed line in the 24 MPa PCP, 7 liter connecting rod is an internal oil supply line to provide pressure lubrication to the piston pin.

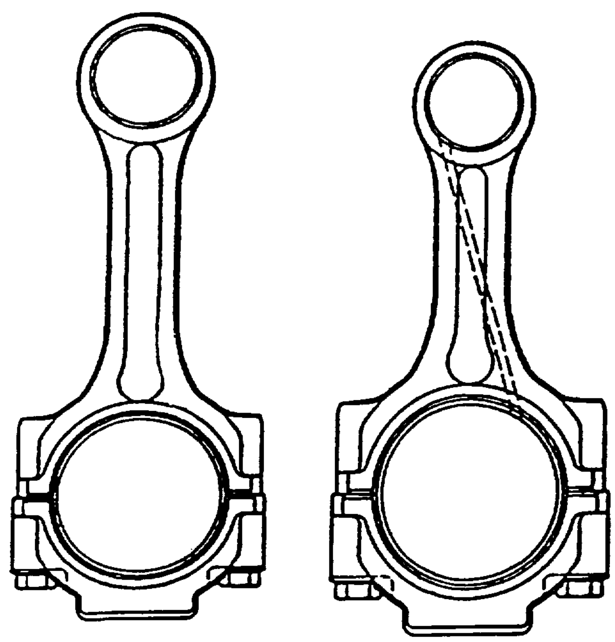


Figure 29 - Connecting Rod Size Comparison

Piston Preliminary Design

Piston heat rejection is predicted to be a significant factor in reducing engine SFC (Table 4). Two piston insulation concepts, thick thermal barrier coating systems and air gap insulation, were included in the preliminary design work. The predicted heat transfer for the air gap and TBC pistons was shown in Table 9.

An articulated piston design was selected to meet the structural requirements imposed by the high cylinder pressures. The piston can be designed using a crown material with high temperature strength. Either an iron or aluminum skirt may be used with the articulated piston design. An aluminum skirt was selected based on finite element model results.

A 3.5 mm. graded coating system applied to a SAE 4140 crown was selected for the TBC piston option. The crown of the TBC piston is very similar to a piston being tested in contract DEN 3-332. Piston finite element model analyses indicated that the metal crown and graded zirconia coating system could meet the piston life goals of 10,000 hours.

An air gap piston design that does not rely on a thermal barrier coating to reduce the piston heat loss was also included in the preliminary engine design and cost analysis. A sketch of the air gap piston geometry is shown in Figure 30. The piston crown is a composite structure that is welded together to form the air gaps between the upper and lower parts. Several welding options will be explored in Phase II of the contract.

Figure 30 illustrates the proposed crown, skirt and piston pin relationship. Provision has been made to supply cooling oil to the back of the the ring belt area and the under side of the piston crown to control the piston ring temperatures and minimize deposits in the ring grooves. The TBC insulated piston has the advantage of not requiring oil cooling behind the ring belt area. The TBC piston does incorporate oil cooling on the under side of the piston crown. Current mineral base lubricants will not be acceptable with either the air gap or TBC piston options due to the high ring belt temperatures.

The piston rings were selected on the basis of current design practices to achieve the sealing and oil control functions. Oil control will be a critical function to meet the anticipated future emission requirements. The final design of the piston rings will be strongly influenced by bench test work and a piston-ring-liner model being developed under this contract.

The piston rings will incorporate a plasma sprayed wear material on the ring faces to meet the engine durability goals. Ring facing materials, cylinder liner bore surface finish and the high temperature lubricant are being evaluated in other DOE sponsored research projects.

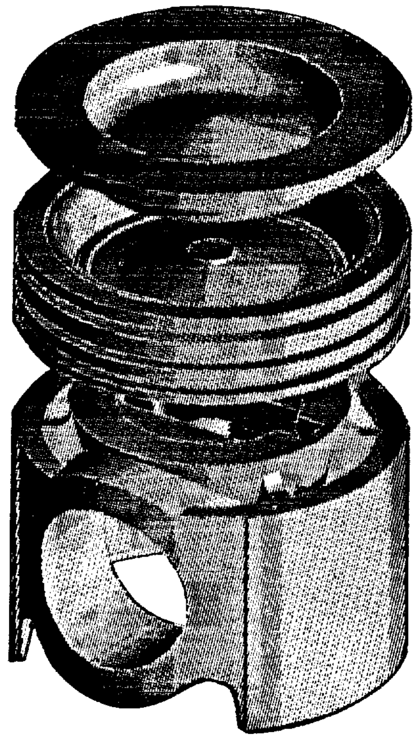


Figure 30 - Welded Air Gap Piston Design

Cylinder Liner Preliminary Design

A 2D FEA was conducted for the 24 MPa PCP cylinder liner. The liner geometry is illustrated in Figure 28. Selective oil cooling will be applied to the top of the cylinder liner. The objective will be to control the liner thermal growth near the top ring reversal location while controlling the liner wall temperature to minimize in-cylinder heat loss and thermal distortion. A cast iron liner will have an acceptable fatigue life and sealing at the 24 MPa cylinder pressure rating.

Cylinder liner bore surface finish will be important for wear, scuffing and oil control at the high in-cylinder operating conditions. A hard wear surface, such as a chrome oxide coating, with a carefully controlled surface texture will be needed. The ring facing material, liner wear surface and high temperature lubricant will be evaluated by bench test during Phase II of the contract.

Cylinder Head Preliminary Design

The cylinder head preliminary design approach was to utilize a gray iron casting with an insert in the bottom deck of the head. The insert forms the top of the combustion chamber and includes the valve seats. The cylinder head construction is illustrated in Figure 31. The head bottom deck insert provides the material strength and the insulation level needed to permit the use of a gray iron casting for the rest of the head.

An insulating air gap has been incorporated in the exhaust ports to minimize the heat loss from the exhaust. Cast-in-place ceramic exhaust port insulation was evaluated during the preliminary design phase but was ruled out due to the space required to achieve the desired level of insulation. Both a metal and a cast ceramic exhaust port liner are feasible.

The cylinder head in Figure 31 will use limited oil cooling. Selective oil cooling will be provided where needed based on the results of a 3D FE model of the head. Selection of the valve stem, valve guide material and cooling oil flows will be based on Phase II bench test results.

Figure 31 illustrates the valve train configuration and the location of the electronic unit fuel injector. The unit injector is cooled by internal fuel flow. The engine cam shaft that mechanically actuates the valves and unit injector is located in a spacer deck between the head and the engine block.

Engine Preliminary Design Dimensions

The crankshaft determines the length and width of the engine block. The connecting rod, piston, head and valve train set the overall height of the engine. The basic engine dimensions, without turbomachinery or air system ducting, for the 7 liter and 10 liter engines are shown in Figure 32. The dimensions for the 10 liter 3176 engine have been included to provide a comparison.

The 10 liter 19 MPa PCP engine is slightly larger than the 10 liter 3176 engine [2] due to the structural loads created by the increased cylinder pressure. The 7 liter, 24 MPa PCP concept engine has 30% less displacement compared with the 3176 engine but has essentially the same dimensions. The decrease in engine displacement, which normally results in a smaller engine, is offset by the need for additional structure required for the 24 MPa PCP operating pressure.

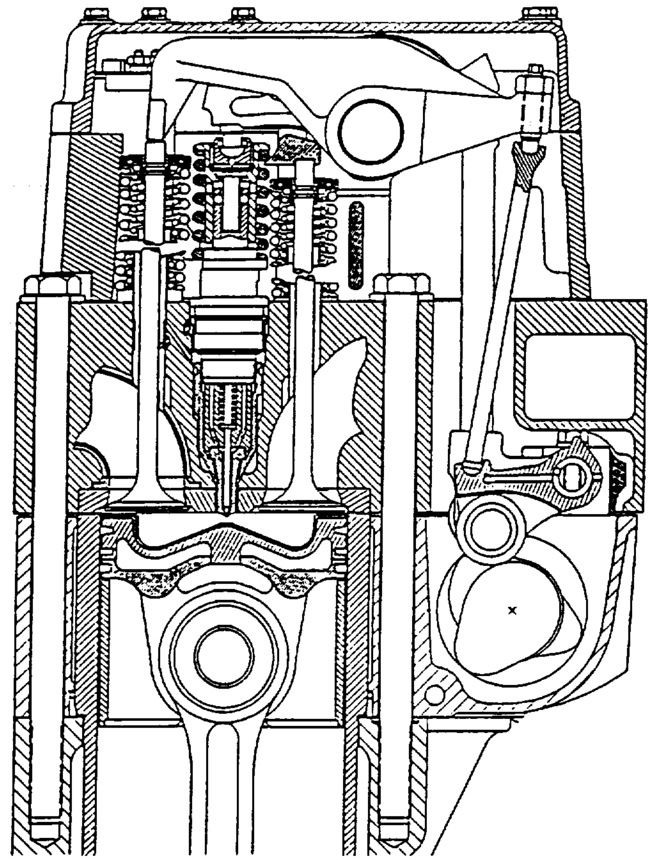
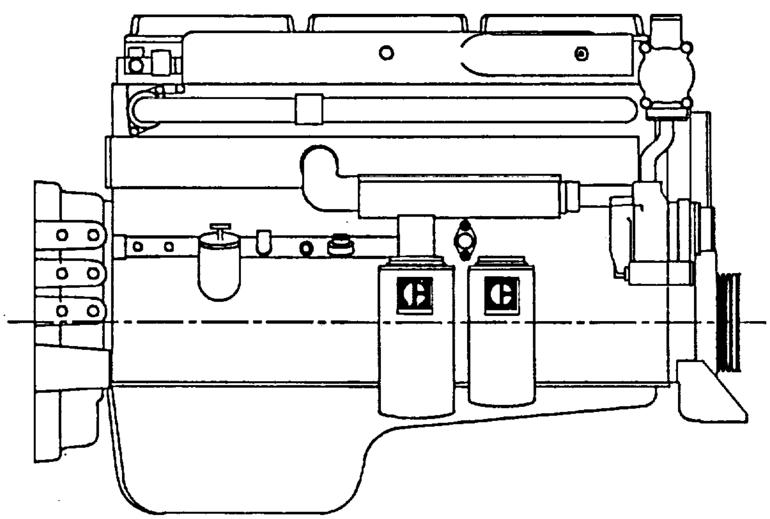


Figure 31 - Cross Section of the Engine Upper Structure



DISPLACEMENT	CYLINDER PRESSURE	LENGTH	WIDTH	HEIGHT
10 Litre	3176 Prod	1295 mm	643 mm	1016 mm
10 Litre	19 MPa	1305 mm	664 mm	1080 mm
7 Litre	24 MPa	1295 mm	650 mm	1016 mm

Figure 32 - Concept Engine Overall Dimensions

Air System Preliminary Design

A preliminary design study was completed to define the air systems for the three concept engines. The 19 MPa PCP engine option incorporated a turbocharger, compound turbine stage and a RBC. The engine configuration used in the costing work was based on a Caterpillar prototype turbocompound engine that is similar in size.[3] The turbomachinery included the necessary ducting and aftercooler.

The 19 MPa PCP concept engine bottoming cycle system was scaled from a Caterpillar prototype system built and tested on a larger diesel engine. Costs for the RBC system were based on the scaled system and included the required heat exchangers, ducting and control system. The 19 MPa PCP engine performance was evaluated with and without the RBC during the Task II economic analysis because the bottoming cycle system initial costs were significant relative to the cost of the 19 MPa PCP engine.

The 22 and 24 MPa PCP engine options incorporate a series turbocharger and a turbocompound stage with gear box. A sketch of the 24 MPa PCP engine with series turbocharger and turbo-compound stage is shown in Figure 33. This configuration was used in preparing the component cost estimates in Task II. Inter-connecting ducting for the turbocharger system, intercooler and aftercooler are not shown in Figure 33 but were included in the engine cost estimates.

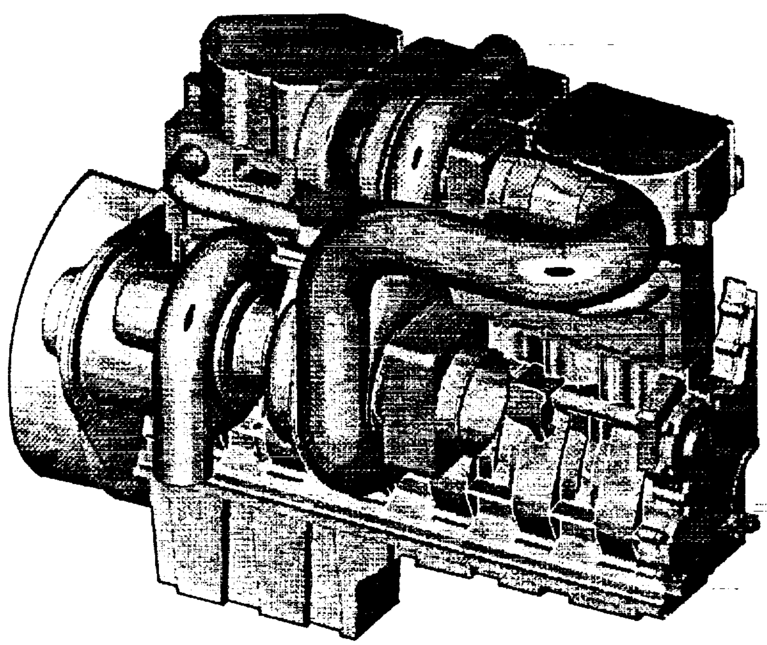


Figure 33 - Series Turbocharger Configuration for 24 MPa Engine

V - CONCEPT SELECTION

The second task objective was to select in-cylinder components and engine systems for cost analyses, technical payoff analyses and technical risk vs. cost tradeoff analyses. Three concept engines from Task 1 were used to evaluate the in-cylinder components. Preliminary component and engine designs from Task 1 provided the basis for estimating the customer costs for each engine option.

The three engine options were modeled using an engine cycle simulation program to predict the expected engine performance. The engine models were used in a truck route simulation model to predict the fuel consumption for two typical truck routes, Chicago, Illinois to Des Moines, Iowa and Salt Lake City to San Francisco. Engine fuel and lubricant costs from the truck route simulation model were used for the economic analyses.

Engine maintenance costs were estimated from current production engine records and were modified to incorporate the anticipated maintenance costs for new in-cylinder components. The engine cost, fuel and lubricant costs and engine maintenance costs were used to predict the customer return on investment (ROI) for each engine option. Figure 34 outlines the economic analysis strategy.

The in-cylinder engine component technical payoff analysis was based on the component contribution to the reduction in engine SFC, component/engine cost and component durability. In-cylinder component contribution to SFC was quantified using the engine cycle simulation model. Component contribution to the life cycle costs were evaluated by estimating the incremental cost of the new component and the expected service life. Cost of engine modifications, such as the crankshaft and block, were also accounted for in the economic analyses.

Technical risk analyses were based on life predictions from finite element models to assess the life and durability of the in-cylinder components. Technical risk vs. technical payoff was also assessed in selecting the in-cylinder components for the economic analyses.

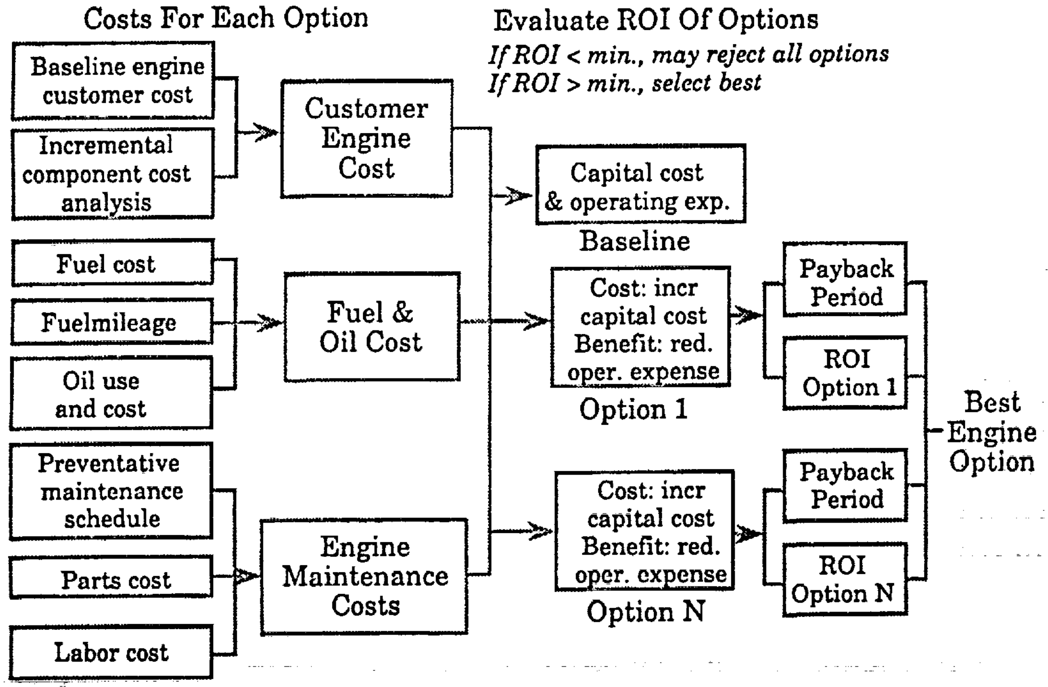


Figure 34 - Cost Analysis Strategy for Engine Options

Engine - Component Cost Analysis

The customer engine cost input to the in-cylinder components cost analysis was based on an incremental component cost analysis. The 3176 engine was selected for the baseline engine customer cost. The 3176 engine is a new production engine that uses state-of-the-art manufacturing technology. The 3176 engine is approximately the same size and power rating as the proposed concept engines.

Incremental component costs were calculated relative to 3176 engine component costs. In some cases, such as the connecting rod, the incremental cost could be determined quite precisely because there was very little difference in the manufacturing operations. The new connecting rods were heavier so material costs were adjusted accordingly.

A matrix of the components evaluated in preparing the incremental costs is shown in Table 13. In the case of the crankshaft for the 7 liter and 10 liter engines, the only change relative to the 3176 crankshaft was the size and material cost. All the manufacturing operations remained the same. In the case of the exhaust manifold, with both a material and a size change, a more complete analysis was required.

TABLE 13 - IN-CYLINDER COMPONENTS MODIFICATIONS

Component	10 Liter Modification Size	10 Liter Modification Mtl	10 Liter New	7 Liter Modification Size	7 Liter Modification Mtl	7 Liter New
Crankshaft	X			X		
Connecting Rod	X			X		
Block	X			X		
Cylinder Head	X			X		
Valves		X			X	
Valve Guide		X			X	
Insulating Insert			X			X
Exhaust Port Liner			X			X
Exhaust Manifold	X	X		X	X	
Piston			X			X
Cylinder Liner		X		X	X	
Air System						
Turbocharger LP	X	X		X	X	
Turbocharger HP						X
Compound Turbine			X			X
Gear Box			X			X
Bottoming Cycle			X			

Several cylinder head machining options were explored to evaluate cylinder head modifications for the bottom deck insert and the exhaust port liners. The manufacturing path selected was to counter bore the head bottom deck to accept the head inserts. This did not represent a large cost penalty because a production head has 4 counter bores per cylinder for individual valve seats. The valve seats are incorporated in the head insert so the only cost penalty is due the added cost of the head insert material.

Several manufacturing studies were completed on the pistons for the concept engines. Both a TBC system on a ferrous articulated piston and a welded air gap articulated piston were analyzed. The incremental cost for the TBC piston was determined for the plasma spray application of the coating system and machining the coating system to a final profile. The costing was based on an automated system to spray the piston, automated transfer of the piston to a machining station, machining the coating and inspecting the finished piston.

Cost estimates for the air gap piston were based on transfer machining and automated welding of the composite piston crown. Inertia welding and laser welding were considered in estimating the cost of joining the piston crown. The piston pin and skirt for both the TBC and air gap piston configurations are similar to current production pistons.

The turbocharger for the 10 liter engine is similar to production turbochargers in size and material. The series turbochargers for 7 liter engines are also similar to production turbochargers. The high pressure turbo compressor may have a material change to meet higher inlet temperatures if an intercooler is not used.

The compound turbine and gear box for both the 7 and 10 liter engines were treated as new components, but the cost estimates were based on experience with prototype systems of the same size. Cost estimates for the Rankine bottoming cycle were extrapolated from the design and cost of a system for a larger engine.

The <u>incremental</u> component costs used to estimate the concept engine customer costs are shown in Table 14. The increased crankshaft cost is due to the additional material required for the crankshaft forgings. The incremental cost for the connecting rods is due to the added material and rifle drilling an oil supply line in the rod. The incremental cost shown in Table 14 is for 6 connecting rods.

Both metal and ceramic material options are shown for the cylinder head inserts, cylinder head exhaust port liners and cylinder liner coatings. The concept engine costs were estimated using both metallic and ceramic in-cylinder components. The thermal barrier coated (TBC) piston was included in the concept ceramic engine configuration. The air gap piston was included in the metal engine configuration.

Estimated customer engine costs used for the engine economic analyses are summarized in Table 15. The 3176 engine, shown in Figure 34, was used for the baseline cost in the economic study. [2] Metal and ceramic options are shown for the 7 liter, 24 MPa PCP engine and the 10 liter, 19 MPa PCP engine. The 10 liter engine system cost includes the cost for the RBC system. The incremental cost for the 7 liter, 22 MPa PCP engine is $160 less than the 24 MPa PCP engine option.

TABLE 14 - CUSTOMER INCREMENTAL COMPONENT COST ESTIMATES

Component		10 L (19MPa)	7L (24MPa)
Crankshaft		201.44	111.92
Con rods		106.00	58.55
Block (casting)		159.00	88.84
Cylinder head - casting		134.79	75.10
Valves (Pyromet)		215.00	120.00
Valve guides (Tribomet 20)		180.00	180.00
Cylinder head inserts*	metal	635.00	492.00
	ceramic	775.00	600.00
Port liner*	IN625	122.00	68.00
	ceramic	107.40	60.00
Exhaust manifold (D5S)		296.85	266.24
Pistons*	TBC	996.00	792.00
	airgap	522.90	415.80
Liner coating*	metal	264.00	180.00
	ceramic	360.00	264.00
Hp turbo			865.00
- housing (D5S)			200.00
Turbocompound - gear set		1200.00	1200.00
- pwr turb stg		890.00	650.00
Camshaft		173.14	96.22
Miscell hdw & piping		430.00	532.00
RBC		14,330.00	

*2 Alternatives Shown

TABLE 15 - ESTIMATED CUSTOMER ENGINE COSTS

3176 (Baseline)	$14,440
7 liter - 24 MPa - metallic	$20,000
7 liter - 24 MPa - ceramic	$20,600
10 liter - 19 MPa - metallic*	$33,850
10 liter - 19 MPa - ceramic*	$34,530

*includes $14,330 Rankine Bottoming Cycle

7 liter - 22 MPa incremental cost is $160 *less* than 24 MPa

Engine Part Load Performance Analysis

Fuel consumption over the life of the engine is the major factor in the owning and operating cost of the truck. Part load fuel consumption maps were generated for the concept engines to calculate the fuel consumption in a typical truck application. The part load fuel consumption maps were used as an input to a truck performance and route simulation program to predict the total fuel consumption for the engine economic analyses.

The description of the engine parameters used in the engine cycle simulation to generate the fuel consumption maps are listed in Table 16. The 10 liter, 19 MPa PCP engine incorporates both a turbocompound system and a RBC system to meet the 152 g/kW-hr target SFC.

TABLE 16 - CONCEPT ENGINES PERFORMANCE PARAMETERS

SFC	152.7	155.0	152.1
Peak cylinder pressure, MPa	19	22	24
RBC	yes	no	no
Turbocompound	yes	yes	yes
Displacement, L	10.3	7	7
Rated speed, rpm	1600	1600	1600
BMEP, MPa (psi)	1.9 (275)	2.8 (406)	3.0 (435)
Turbochargers (pressure ratio)	single (3.03)	series (4.3)	series (4.7)
Turbocharger efficiency, %	69	74	76
Compression ratio	17	16	16.5
Insulation	Uncooled head with ceramic insert Oil cooled liner Low heat rejection piston		

A 3D bar graph of the expected part load performance of the concept engines is shown in Figure 35. Engine SFC is shown at 100% and 50% of rated power at the rated speed of 1600 rpm and peak toque speed of 1050 rpm. A road load SFC that represents a typical truck cruising speed and power requirement, was also included to define the concept engines fuel maps.

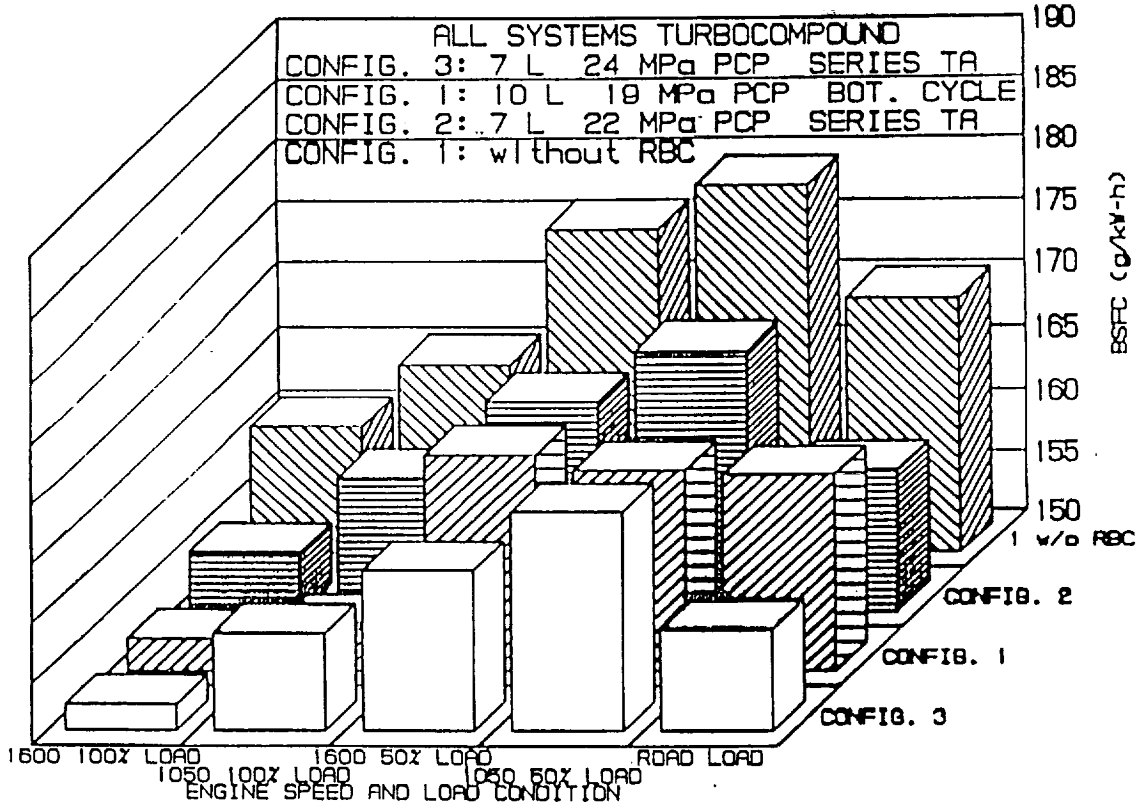


Figure 35 - Concept Engines Part Load SFC

The part load SFC for the 10 liter, 19 MPa PCP engine is shown with and without the bottoming cycle system. The SFC of the 10 liter engine without the bottoming cycle is the maximum SFC that would be expected if the bottoming cycle was ineffective during transient operation of the engine.

The predicted part load performance for the concept engines is compared to the 3176 reference production engine in Table 17. The technical and economic payoff resulting from the reduced SFC of the concept engines was determined by calculating the annual fuel consumption for the baseline and concept engines.

The engine economic analysis process is outlined in Figure 36. The engine part load fuel consumption data was used as input to a truck performance and route simulation program. Engine fuel and lubricant consumption, initial engine cost and engine maintenance and overhaul costs are required inputs for the engine investment analysis program.

TABLE 17 - PREDICTED CONCEPT ENGINES SFC (g/kW-hr)

Engine	Reference Production Engine	ICC Concept Engines		
		19MPa (RBC)	22 MPa	24MPa
Rated Speed, Full Load	190.9	152.7	155.1	152.2
Rated Speed, Half Load	203.7	168.1	167.5	163.3
Peak Torque Speed, Full Load	191.8	154.0	161.3	158.0
Peak Torque Speed, 1/2 Load	197.7	166.8	171.6	168.2
Road Load	194.9	166.4	162.0	158.4

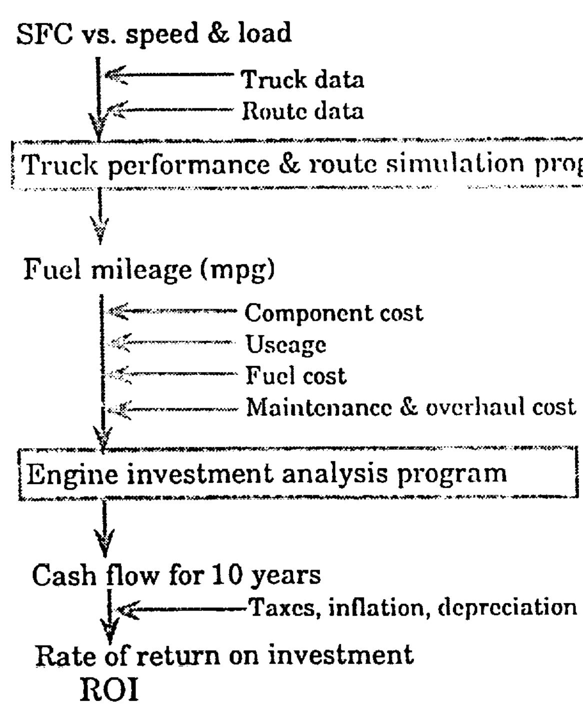


Figure 36 - Concept Engine Economic Analysis Outline

Engine Exhaust Emissions

Control of the diesel engine exhaust emissions will have an impact on the engine fuel consumption. The concept engines SFC values shown in Figure 35 and Table 17 represent a fuel injection timing for best fuel consumption. Diesel engine technology to meet the year 2000 gaseous emissions goals will have to demonstrate at least a 5 g/hp-hr NO_x and 0.1 g/hp-hr particulate, or lower levels. Reducing NO_x and particulate level to meet the future goals will be the principal technical challenge. Current diesel engine gaseous hydrocarbon (HC) and CO_2 emissions are already within the anticipated limits.

A Caterpillar model was used to estimate the NO_x emissions from the three concept engines at a fuel injection timing for best SFC. The model was then used to estimate the increase in specific fuel consumption when fuel injection timing was retarded to meet a 4.5 g/hp-hr NO_x emissions level. The Caterpillar model has been correlated with measured engine emissions.

The concept engine NO_x levels expected at a fuel injection timing for best SFC are shown in Figure 37. The 10 liter, 19 MPa PCP engine has the highest predicted NO_x level due to an earlier fuel injection timing and a higher compression ratio. The expected increase in SFC due to retarding the fuel injection timing to a 4.5 g/hp-hr NO_x level is shown in Figure 38.

Figures 37 and 38 provide an estimate of the increase in SFC at rated power if retarded fuel injection timing is used to lower the engine NO_x level. Engine emissions are certified on the EPA transient heavy duty engine cycle so the steady state emissions do not correspond on an absolute basis. The SFC estimates do provide a basis for comparing the engine system concepts.

A predictive technique has not been developed for particulate emissions for large changes in engine design and operating conditions. Only the trend of particulate emissions for the concept engines can be discussed.

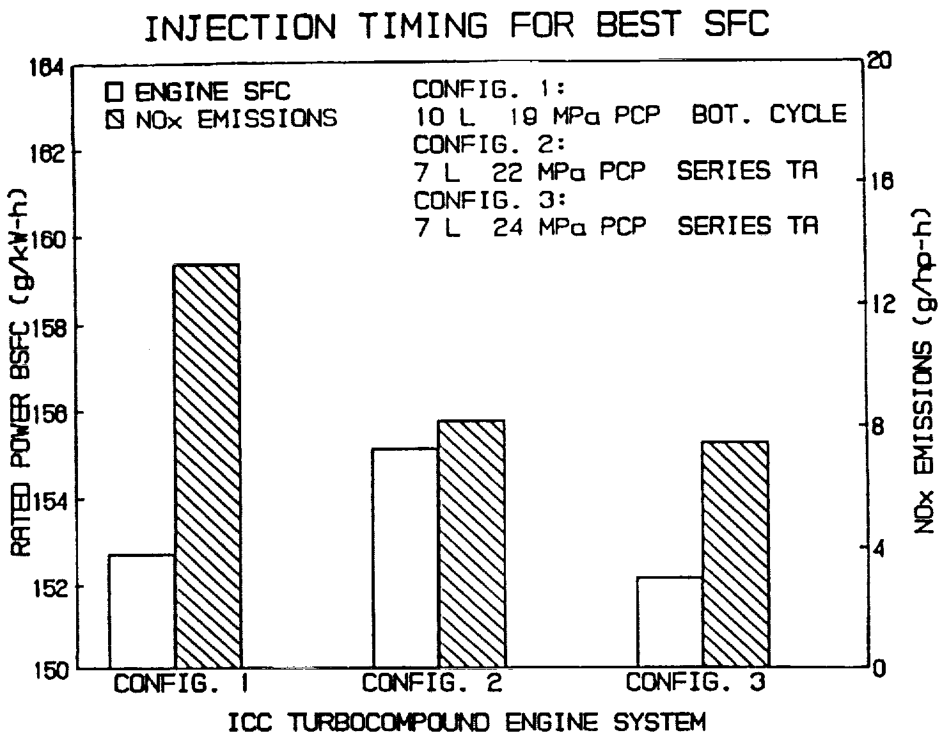


Figure 37 - Rated Power SFC and NO_x Emissions for Best SFC

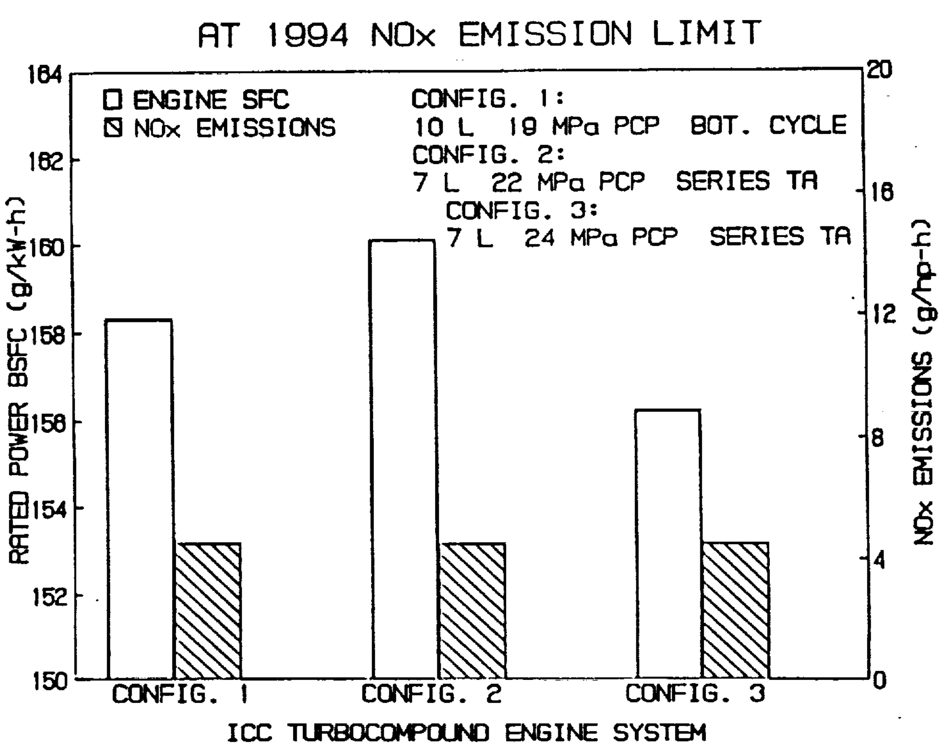


Figure 38 - Rated Power SFC and NO_x Emissions at 4.5 g/hp-hr

Some of the diesel engine development strategies that are being used to control particulate and gaseous emissions include:

1) High pressure fuel injection,
2) Low sulfur fuel,
3) Exhaust after-treatment,
4) Combustion chamber geometry,
5) Oil control.

The proposed concept engines are compatible with any of these control strategies. Reliable data on the trend of soot and particulate generation in low heat rejection engines is still very limited, but the trend seems to indicate that low heat rejection engines should have a lower soot production. The use of low sulfur fuel and higher pressure fuel injection systems tend to reduce particulate emissions.

Oil cooling of the cylinder liner is proposed in the concept engines to control liner temperatures and oil viscosity. This should minimize the oil contribution to the soluble organic fraction (SOF) of the particulates. The design of the piston ring pack for sealing and oil control at the proposed operating conditions will be a significant factor in meeting the particulate emissions goal. A ring pack dynamics model is being developed to assist in the piston ring design and development.

Engine Maintenance Costs

Engine maintenance costs are a significant factor in the owning and operating costs of heavy duty truck. Experience has shown that following a preventive maintenance schedule will minimize engine maintenance costs. A preventive maintenance schedule, based on Caterpillar truck engine experience, was used to estimate maintenance costs for the concept engines.[4] Table 18 shows the maintenance schedule used in the analyses.

Three maintenance levels were use in the cost analysis. The maintenance performed at each level is shown in Table 18. An oil change interval of 25000 miles (approximately 500 hours) was selected based on the use of a synthetic lubricant and a properly sized oil sump. An in-frame overhaul of the engine is scheduled at 500,000 miles (approximately 5 years). The useful life of the engine was assumed to be one million miles.

TABLE 18 - ENGINE PREVENTIVE MAINTENANCE SCHEDULE

PREVENTIVE MAINTENANCE SUMMARY	FILE = ICCPM1.DAT	DATE:	
SCHEDULED MAINTENANCE ITEM INTERVAL -- Miles ---->	PM #1 25000	PM #2 100,000	PM #3 200,000
REPLACE			
ENGINE OIL	▬	▬	▬
OIL FILTER	▬	▬	▬
FUEL FILTER	▬	▬	▬
AIR FILTER		▬	▬
COOLANT OIL FILTER	▬	▬	▬
OIL TEMPERATURE REGULATOR		▬	▬
INSPECT			
HEAT EXCHANGER FINS	▬	▬	▬
BATTERY ELECTROLYTE LEVEL	▬	▬	▬
BELTS	▬	▬	▬
TURBOCHARGER		▬	▬
ENGINE MOUNTS		▬	▬
DAMPER		▬	▬
VALVE ROTATORS		▬	▬
TEST FUEL INJECTORS		▬	▬
CLEAN			
BATTERY TERMINALS	▬	▬	▬
CRANKCASE BREATHER	▬	▬	▬
PRIMARY FUEL FILTER	▬	▬	▬
OBTAIN S.O.S. SAMPLE	▬	▬	▬
CHECK/ADJUST			
SET POINT		▬	▬
VALVE LASH		▬	▬
AIR-FUEL RATIO CONTROL		▬	▬
LUBRICATE FAN DRIVE	▬	▬	▬
DRAIN WATER FROM AIR TANK	▬	▬	▬
STEAM CLEAN ENGINE			▬
TEST RUN / CHECK FOR LEAKS	▬	▬	▬

Engine Operating Costs

Engine operating costs consist of the fuel that is burned, maintenance costs and overhaul costs. Maintenance costs were based on recommended schedules shown in Table 18. Maintenance costs were adjusted for the components unique to the low heat rejection concept engines. For instance, maintenance on the water/glycol coolant system was eliminated. Maintenance costs for the second turbocharger, compound turbine and gear box were added.

Fuel burned represents the largest single operating cost. One of the challenges was to provide a realistic estimate of the amount of fuel that an engine would consume over a ten year life or one million miles. A truck engine performance and route simulation program was used to calculate the engine fuel usage (Figure 36). Two truck routes, Chicago, Illinois to Des Moines, Iowa and Salt Lake City to San Francisco, were selected to evaluate the fuel consumption. The Chicago to Des Moines route is representative of a fairly level truck route encountered in the Midwest. The Salt Lake City to San Francisco route includes some of the steepest grades in the Interstate highway system.

The truck route simulation was first run with a 3176 engine to provide the baseline fuel consumption. The results of a Salt Lake City to San Francisco run are shown in Table 19. The validity of the truck route simulation has been verified by actual truck mileage tests. The truck routes were then run with the three concept engines at the same gross vehicle weight.

TABLE 19 - SALT LAKE TO SAN FRANCISCO 3176 ENGINE EVALUATION

Distance	699 miles
Average Road Speed	59 mile per hour
Total Driving Time	11.9 hours
Total Number of Gear Shifts	445
Total Fuel Consumption	112 gallons
Load Factor	65%
Time at Rated Power	37.6%
Average Fuel Consumption	6.24 mpg

The results of the truck simulation with an 80,000 pound gross vehicle weight for the two routes are summarized in Table 20. The Midwestern route produced the highest average speed and the best mile-per-gallon (mpg) performance. The steep grades, both favorable and adverse, are reflected in the Salt Lake to San Francisco and return route. The Salt Lake City to San Francisco route has a higher average mpg because there are more favorable grades than adverse grades in that direction on Interstate 80. The 3176 engine fuel consumption based on these two routes was used as a baseline for comparison with the concept engines in the same truck configuration.

TABLE 20 - TRUCK ROUTE SIMULATION MILEAGE RESULTS

3176 Road Mileage Results

Route	Chicago to Des Moines	Salt Lake City to San Francisco	San Francisco to Salt Lake City
Distance, miles	328.4	698.9	698.9
Average Speed, mph	62.93	58.78	57.20
Mileage, mpg	6.40	6.24	5.82
Load Factor, %	67.8	65.0	67.8

The truck route fuel consumption values were used as the basis for the comparing the concept engines with the 3176 truck engine. The fuel consumption values are based on injection timing for best fuel consumption. No attempt was made to modify the engine fuel consumption maps to anticipate changes required to meet future emission regulations.

Approximately 50% of the truck operation is near the road load power point shown in Figure 35 and Table 9. The fuel consumption and mpg differences between engines are similar to the differences in road load SFC. The 7 liter, 24 MPa PCP concept engine has 25% lower fuel consumption compared to the baseline 3176 engine. The 22 MPa PCP, 7 liter engine has a slightly higher road load SFC which was reflected in a higher annual fuel consumption.

VI - CONCEPT ENGINES ECONOMIC ANALYSES

The acceptance of new engine technology in the on-highway truck market will determined by engine reliability and economic payback. A customer cost analysis of owning and operating costs for the concept engines was completed using a Caterpillar 3176 truck engine as the baseline case. The engine cost analysis strategy is shown in Figure 39. Data required for the engine cost analysis, customer engine cost, fuel and lubricant costs and engine maintenance costs, were developed in Task 2. Assumptions used in the engine economic analyses are listed in Table 21.

TABLE 21 - ASSUMPTIONS FOR CUSTOMER TRUCK COST ANALYSIS

Study Period	10 years	1 million miles
Investment	incremental customer	based on accounting data and suppliers' estimates
Operating Expenses	fuel, lube, maintenance and engine overhaul	based on engine and truck operating data
Contingency	10%	applied to investment and operating expenses
Depreciation	3 year property	1986 Tax Act
Inflation	4%	long term projection
Income tax	38%	includes state, US and other taxes.

The incremental customer cost for new engine technology can be a significant factor in deciding whether to invest in new engine concepts. Engine availability, reliability and durability are also important factors in customer acceptance. The truck owner recognizes that fuel consumption is the largest single factor in the owning and operating costs. Diesel fuel costs have experienced large fluctuations and this is expected to continue. Higher fuel costs will accelerate the demand for new engine technology that reduces fuel consumption, if the customer is convinced that reliability and durability are incorporated in the new engines.

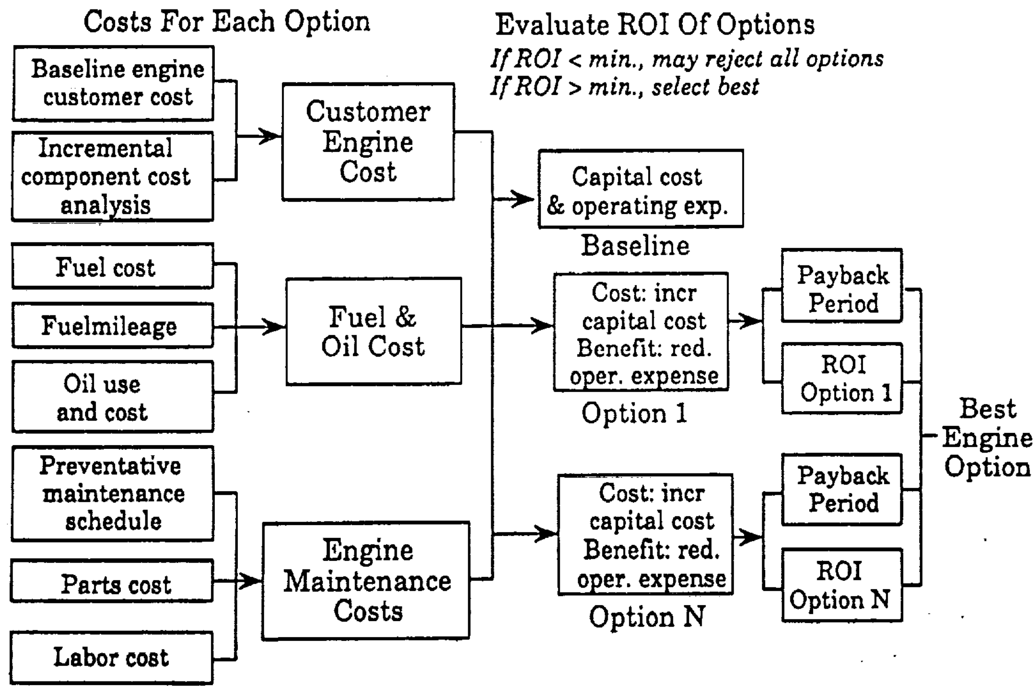


Figure 39 - Engine Economic Cost Analysis Strategy

The engine cost analyses were first run assuming that diesel fuel was available at $1.00 per gallon. A commercial CD grade heavy duty lubricant at $1.00 per quart was assumed for the baseline 3176 engine analysis. A high temperature synthetic lubricant at $5.00 per quart was assumed for the three concept engines.

The incremental return-on-investment for the three concept engines using 10 year engine life and $1.00 per gallon fuel is shown in Table 22. The 10 liter, 19 MPa PCP engine concept does not have a favorable rate of return for fuel at $1.00 per gallon. The higher initial incremental engine cost is barely offset by the reduced fuel consumption.

TABLE 22 - CONCEPT ENGINES COST ANALYSIS - $1.00 GAL. FUEL

	Rated SFC	ROI
3176	191 g/kW-h	Base
19 MPa - 10 liter w/RBC	154 g/kW-h	2%
22 MPa - 7 liter	155 g/kW-h	32%
24 MPa - 7 liter	152 g/kW-h	36%

Assumptions: 10 year life
Fuel $1.00/gallon

Concept Engine Sensitivity to Fuel Cost

The cost of diesel fuel is subject to a number of economic and political pressures. Long term projections of world oil supplies suggest that the petroleum reserves are limited and the long term trend will be upward in response to supply and demand pressures. Taxes can also have a significant impact on fuel cost. Political disruptions of the fuel supply have and will probably continue to cause large fluctuations in the cost of fuel.

A sensitivity analysis to fuel costs was run for the three concept engines to evaluate the impact of fuel cost on the ROI. The effect of fuel costs from $0.50 to $2.00 per gallon on the concept engines' return-on-investment is shown in Figure 40.

Doubling the cost of diesel fuel to $2.00 per gallon begins to make the 10 liter, 19 MPa PCP concept engine with a RBC system start to look economically feasible. However, at $2.00 per gallon, the 7 liter engines are much more attractive. The initial investment cost of the bottoming cycle system for a truck engine does not appear to present an attractive alternative. The low heat rejection, high cylinder pressure, turbocompound engine will be economically attractive when the price of diesel fuel increases above $1.00 per gallon if the reliability and durability goals of the engine concept are demonstrated.

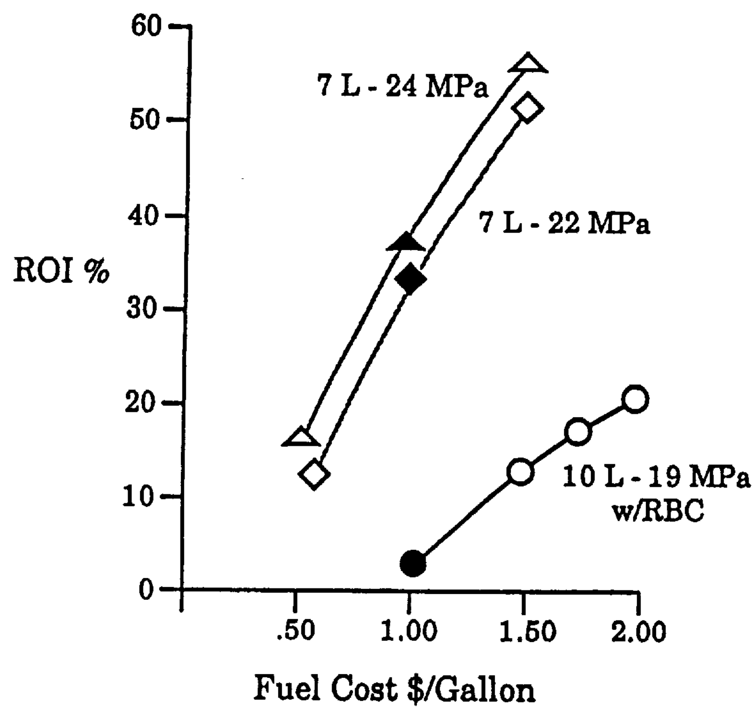


Figure 40 - ROI Sensitivity to Fuel Costs

Both ceramic and metal in-cylinder components were included in the concept engine preliminary design and analysis (see Table 14). Use of ceramic in-cylinder components resulted in a lower predicted SFC due to the lower heat loss from the combustion chamber and higher energy recovery by the turbocompound system. The ceramic engine configuration incorporated thermal barrier coated pistons, ceramic cylinder head inserts and ceramic coatings on the cylinder liners. Incorporating the ceramic in-cylinder components did result in a higher initial engine cost.

The trade-off between ceramic and metal in-cylinder components is shown in Table 23. At $1.00 per gallon, the metal engine had a higher ROI due to the lower initial cost. The metal and ceramic components were expected to have equal life so there was no maintenance or overhaul cost penalty. At $1.10 per gallon fuel cost the returns-on-investment were equal. At higher fuel costs, the ceramic insulated engine should have an economic advantage. Obviously, the results of the ceramic vs. metal in-cylinder components is too close to call based on this study. Both types of component insulation schemes are recommended for further evaluation.

TABLE 23 - CERAMIC vs. METAL INSULATION ROI TRADE-OFF

24 MPa - Liter Engine	Rated SFC	ROI
Ceramic insulation	152	38%
Metal insulation	153	40%

Fuel $1.00/gallon

Equal ROI @ fuel = $1.10/gallon

ROI Sensitivity to Initial Engine Cost

All the engine economic studies have shown that the initial customer cost of the engine will have a significant effect on the ROI. Initial engine cost can also have a significant impact on customer acceptance, particularly if the ROI is marginal.

Figure 41 demonstrates the impact of the <u>incremental</u> engine cost on the predicted ROI with a fuel cost of $1.00 per gallon. The 10 liter, 19 MPa PCP engine initial cost would have to be reduced by at least $10,000 to bring the bottoming cycle engine into the range of economic interest. The cost reduction for the 19 MPa PCP bottoming cycle engine could be less as the price of fuel increased above $1.00 per gallon but this still would not be as economically attractive as the 7 liter, 24 MPa PCP engine configuration.

The incremental engine cost sensitivity of the 7 liter engines is also shown in Figure 41. The initial cost of the 7 liter engines could increase by as much as $10,000, with fuel costs at $1.00 per gallon before the ROI becomes unattractive. Estimated engine costs for the study were based on the incremental costs of the new in-cylinder and engine components relative to a production base-line engine. The incremental cost sensitivity is sufficiently great to permit selection of the 7 liter engine configuration for additional evaluation in Phase 2 of the contract.

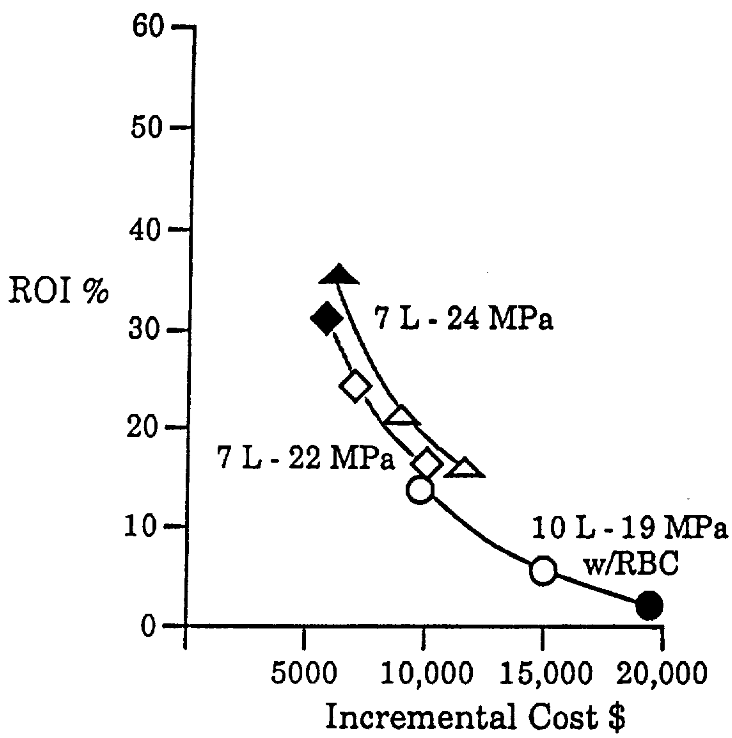


Figure 41 - ROI Sensitivity to Incremental Engine Cost

Concept Engine Summary

An engine configuration and a path to approach the 152 g/kW-hr SFC goal have been identified. The economics of the engine approach look favorable based on the preliminary design and cost study. Key elements in reaching the fuel consumption goal are:

1. Higher peak cylinder pressures - 22 to 24 MPa
2. Higher BMEP - 2.8 to 3.0 MPa
3. Higher air system boost - series turbochargers
4. Higher turbomachinery efficiency with intercooling and aftercooling. Efficiency - 74% to 76%
5. Oil cooled - low heat rejection components
6. Turbocompounding for exhaust energy recovery

The impact of future emission regulations on the engine SFC cannot be predicted with accuracy. Control of NO_x and particulates are key technical issues.

NO_x generation in the concept engines will probably be higher than current engines due to the higher in-cylinder temperatures. One strategy is to retard fuel injection timing to lower the NO_x formation with some increase in fuel consumption. The upper bounds on the increase in fuel consumption by retarding fuel injection timing were shown in Figure 38. Combustion development at the higher peak cylinder pressures will be required to confirm this projection. The fuel consumption penalty is the smallest for the 7 liter, 24 MPa PCP engine configuration. It is possible that retarding fuel injection timing may not be required if an external NO_x control scheme is adopted.

Particulate regulations will become more stringent for future diesel engines. Incorporating insulated components that raise the in-cylinder temperatures will probably be favorable for reduced soot generation which is a large contributor to the particulate emissions. The other key source of particulate emissions is the soluble organic fraction contributed by the lubricant getting into the combustion chamber. Design of piston ring packs to provide sealing and oil control at the higher cylinder pressures and temperatures will be a key element in qualifying the engines to meet future emissions regulations.

Component design for durability and reliability at the higher peak cylinder pressures and operating temperatures is also a key requirement for the concept engines. Engine and component durability will have to meet or exceed current levels. Key design technical issues include:

1. Cylinder head structural capability at the elevated pressures and temperatures. Minimum head cooling was used in conducting this study.
2. Durable piston insulation. Both ceramic and air gap insulation options should be evaluated by bench and engine testing.
3. Piston ring and liner performance and durability. The piston ring, liner and lubricant development will be key issues in meeting performance, emission and durability goals.
4. Tribology issues. Engine bearing, piston ring and valve guide lubrication and wear will be strongly influenced by the choice of the high temperature lubricant.

7.0 CONCLUSIONS AND RECOMMENDATIONS

A low heat rejection engine configuration designed for higher peak cylinder pressures and power density (higher BMEP) has the potential to meet the SFC objectives. The design and bench test evaluation of the in-cylinder components is the next step to validate the design concept. Both bench tests and single cylinder engine evaluation of the components are recommended. Bench tests are needed to screen candidate materials and lubricants. Engine tests provide the most cost effective method of evaluating the in-cylinder components at the design operating conditions. Specific in-cylinder components that should be tested at the 22 to 24 MPa PCP operating conditions include:

1. Metal and ceramic inserts in a selectively oil cooled cylinder head for structural and thermal capability.
2. Thermal barrier coated pistons and piston air gap insulation to minimize piston heat loss and ring operating temperatures.
3. Cylinder liner coatings, piston ring facing materials and high temperature lubricants to minimize wear and scuffing.
4. Valve and valve guide materials and high temperature lubricant for wear and scuffing resistance.
5. Engine bearing systems for operation at the higher cylinder pressure loads with the high temperature lubricant.

Design and development of the piston, piston rings and cylinder liner will be a key element in meeting engine performance (sealing), durability and emissions goals. Development of the piston-ring-liner model to support this effort is recommended in addition to the component bench testing.

VII ACKNOWLEDGEMENTS

The authors would like to acknowledge the assistance of the Caterpillar team that participated in the Phase I work. The team was composed of members from Power Plant Research, Advanced Materials Division, Engine Division Engineering and Engine Division Manufacturing. Contributions by outside consultants was acknowleged in the body of the report.

The technical and financial assistance of the DOE/NASA Heavy Duty Transport Program supported this effort through Contract DEN 3-374. The assistance and support of NASA progam managers James Wood and Joe Notardonato is gratefully acknowledged.

VIII REFERENCES

1. R.A. Cemenska, "A Rankine Bottoming Cycle System for Heavy Duty Applications", ASME New Engine Technology for Cogeneration, ICE-Volume 2, October 4, 1987

2. J.E. Sibley, "The Caterpillar 3176 Heavy Duty Diesel Engine", SAE 881856, February 1988.

3. D.E. Wilson, "The Design of a Low Specific Fuel Consumption Turbocompound Engine", SAE 860072, February 1986.

4. Operations and Maintenance Manual - 3176 ATAAC Truck Engine, Caterpillar Publication SEBU6163-01, March 1989.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY *(Leave blank)*	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	March 1992	**Final Contractor Report**

4. TITLE AND SUBTITLE

Development of Advanced High Temperature In–Cylinder Components and Tribological Systems for Low Heat Rejection Diesel Engines Phase I – Final Report

5. FUNDING NUMBERS

6. AUTHOR(S)

C. A. Kroeger
H. J. Larson

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

Caterpillar Inc.
100 N.E. Adams Street
Peoria, IL 61630

8. PERFORMING ORGANIZATION REPORT NUMBER

None

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

U.S. Department of Energy
Office of Propulsion Systems
Washington, D.C. 20585

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

DOE/NASA/0374–1
NASA CR–187158

11. SUPPLEMENTARY NOTES

Prepaid under Interagency Agreement DEA10191CE50306
Project Manager: J. J. Notardonato, NASA Lewis Research Center, Cleveland, OH 44135

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Unclassified – Unlimited
Subject Category 85
DOE Category UC–96

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

Analysis and concept design work completed in Phase I have identified a low heat rejection engine configuration with the potential to meet the Heavy Duty Transport Technology program specific fuel consumption goal of 152 g/kW–hr. The proposed engine configuration incorporates low heat rejection, in–cylinder components designed for operation at 24 MPa peak cylinder pressure. Water cooling is eliminated by selective oil cooling of the components. A high temperature lubricant will be required due to increased in–cylinder operating temperatures. A two–stage turbocharger air system with intercooling and aftercooling was selected to meet engine boost and BMEP requirements. A turbocompound turbine stage is incorporated for exhaust energy recovery. The concept engine cost was estimated to be 43% higher compared to a Caterpillar 3176 engine. The higher initial engine cost is predicted to be offset by reduced operating costs due to the lower fuel consumption.

14. SUBJECT TERMS	15. NUMBER OF PAGES
Low Heat Rejection Diesel Engine	**70**
Ceramic Components	**16. PRICE CODE**
Turbocompounding	**A05**

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	**Unclassified**	**Unclassified**	